1.0

4.5

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NAVAL POSTGRADUATE SCHOOL
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LEVEL

THESIS

DESIGN MODEL FOR THE HEAT TRANSFER
IN A SHORT STRAIGHT TUBE BOILER

by

Leo W. Vollmer, Jr.

June 1981

Thesis Advisor: P.F. Pucci

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Design Model for the Heat Transfer
in a Short Straight Tube Boiler

by

Leo W. Vollmer, Jr.
Lieutenant, United States Navy
B.S.E.E. Purdue University, 1974

Submitted in partial fulfillment of the
requirements for the degree of

MASTER OF SCIENCE IN MECHANICAL ENGINEERING

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ABSTRACT

A design model for the Short Straight Tube Boiler with a segmented fin-tube arrangement was developed. This model was integrated for a single tube and applied in a computer program written in BASIC for the Hewlett Packard 9845 model B desk-top computer.

Water-side Reynolds numbers were varied in order to investigate the performance of this boiler. For an overall tube length of 39.4 inches, a Reynolds number of 840 (29.65 lbm/hr) resulted in obtaining 50° superheat for an operating pressure of 600 psig. With these conditions, saturated boiling begins at 11.45 inches and superheating at 21.88 along the tube length.

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NOMENCLATURE

English Letter Symbols

A	-	Area (FT^2)
A_b	-	Frontal Area Blocked by Tubes and Fins (FT^2)
A_{bt}	-	Bare Tube Area (FT^2)
A_f	-	Heat Exchanger Frontal Area (FT^2)
A_{fin}	-	Fin Area (FT^2)
A_{ff}	-	Cross-Sectional Area for Fluid Flow (FT^2)
A_{min}	-	Minimum Cross-Sectional Area for Gas Flow (FT^2)
A_{si}	-	Inside Heat Transfer Area (FT^2)
A_{so}	-	Outside Heat Transfer Area
C_{max}	-	Maximum Heat Capacity (BTU/Hr-F)
C_{min}	-	Minimum Heat Capacity (BTU/Hr-F)
C_{pf}	-	Specific Heat of Water/Steam (BTU/lbm-F)
C_{pg}	-	Specific Heat of Gas (BTU/lbm-F)
d_f	-	Fin Outside Diameter (FT)
d_i	-	Inside Tube Diameter (FT)
d_o	-	Outside Tube Diameter (FT)
D_p	-	Pin Outside Diameter (FT)
f'	-	Friction Factor
F	-	Reynolds Number Factor
G_{max}	-	Maximum Gas Flow Rate Per Square Foot (lbm/hr-FT^2)
h	-	Heat Transfer Coefficient (BTU/Hr-FT^2°F)
H_{ps}	-	Horizontal Pin Spacing (FT)

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I	-	Enthalpy (BTU/lbm)
I_f	-	Enthalpy of Saturated Water (BTU/lbm)
I_{fs}	-	Enthalpy of Vaporization (BTU/lbm)
j	-	Heat Transfer Colburn j-Factor
k_g	-	Thermal Conductivity of Gas (BTU-Hr-FT-F)
ℓ	-	Fin Height (FT)
ℓ_c	-	Length of Cut from Fin Tip (FT)
L_p	-	Average Pin Length (FT)
L_T	-	Tube Length (FT)
N, NTU	-	Number of Transfer Units
N_f	-	Number of Fins Per Inch
N_s	-	Number of Segments in One Fin
$\emptyset_{ONB}$	-	Heat Flux to Initiate Boiling (BTU/Hr-FT^2)
P	-	Pressure (psia)
Q	-	Heat Transfer Rate (BTU/Hr)
R_i	-	Heat Exchanger Inside Resistance (Hr-FT^2°F/BTU)
R_o	-	Heat Exchanger Outside Resistance (Hr-FT^2°F/BTU)
R_{th}	-	Thermal Resistance (Hr-FT^2-F/BTU)
T_f	-	Fin Thickness (FT)
T_g	-	Gas Temperature (°F)
T_{sat}	-	Temperature of Saturated Water (F)
T_{TO}	-	Outside Tube Wall Temperature (°F)
T_{Ti}	-	Inside Tube Wall Temperature (°F)
U_{oi}, U_{oo}	-	Overall Heat Transfer Coefficient (BTU/Hr-FT^2-F)
V_{ps}	-	Vertical Pin Spacing (FT)

x	-	Steam Quality
x_e	-	Equilibrium Quality
x_T	-	True Quality
X_{TT}	-	Martinelli Parameter

Dimensionless Groups

Nu	-	Nusselt Number
Pr	-	Prandtl Number
Re	-	Reynolds Number
Re_{TP}	-	Two-Phase Reynolds Number
St	-	Stanton Number

Greek Letter Symbols

σ	-	Water Surface Tension (lbf/FT)
α	-	Void Fraction
ΔP	-	Pressure Change (psia)
ΔT	-	Temperature Change (F)
ϵ	-	Effectiveness
μ_b	-	Viscosity at Bulk Temperature (lbm/FT-Hr)
μ_w	-	Viscosity at Tube Wall Temperature (lbm/FT-Hr)
η	-	Fin Efficiency
η_s	-	Surface Efficiency
ρ	-	Density (lbm/FT^3)
ρ_ℓ	-	Density of Saturated Water (lbm/FT^3)
ρ_v	-	Density of Saturated Vapor (lbm/FT^3)

I. INTRODUCTION

A. BOILER

1. An Overview

A boiler is an apparatus in the form of a closed vessel constructed for the continuous generation of vapor under pressure through the transference of heat to the liquid which is contained in it. The energy for most boilers is provided by the combustion of the fossil fuels - coal, oil, coke, or gas. Coal is the major fuel, but most boilers are convertible from one fuel to another. Other fuels such as wood, waste gases from industrial processes, and solid wastes such as bagasse (from sugarcane), sawdust, and even trash and garbage serve as energy sources. An increasing number of large steam plants built since 1960 for generating electricity are designed to use nuclear fuel, which provides heat from nuclear fission.

The simple term "boiler" ordinarily refers to the steam boiler which utilizes available heat energy to convert water into steam. Because a boiler produces saturated steam only, it must be distinguished from the steam generator which may include a superheater, economizer, and air preheater as integral and necessary parts of the equipment. In their simplest form, boilers are closed caldrons of water placed over an open fire. Boilers range in size and function from

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the compact units in domestic heating systems to 20-story complexes that drive giant steam turbines for electrical power production. They are built in many forms with rated steaming capacities from 40 lbm/hr to 1,000,000 lbm/hr steam production and operating pressures ranging from 2 psia to the critical pressure of water, 3208 psia.

2. History

The development of the boiler was initiated and has been sustained by the development of the steam engine and steam turbine. Early versions of both, more toys than serious inventions, are referred to in the writings of Hero of Alexandria during the third century B.C. Contrary to popular belief, the principles used in boiler design are not recent developments but are of ancient origin. As early as 200 B.C., boilers were employed for warming water, heating, and household services. An example of this type of boiler is shown in Fig. 1. In this boiler, found in the ruins of Pompeii, the grate consisted of bronze water-cooled tubes and a water-cooled furnace chamber wall. The boiler was internally fired, a feature which did not appear again until the early 18th century. Water-cooled furnaces and water tubes also were innovations not incorporated in boiler design until recent.

However, basic principles for boiler construction, comparable to those set down in 1769 by James Watt for the steam engine, were not widely known until the 19th century. Beginning with Hero, the caldron form of boiler was used.
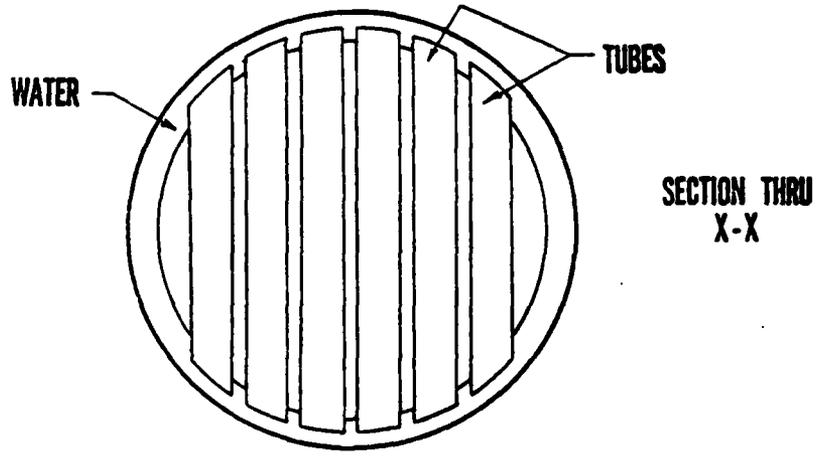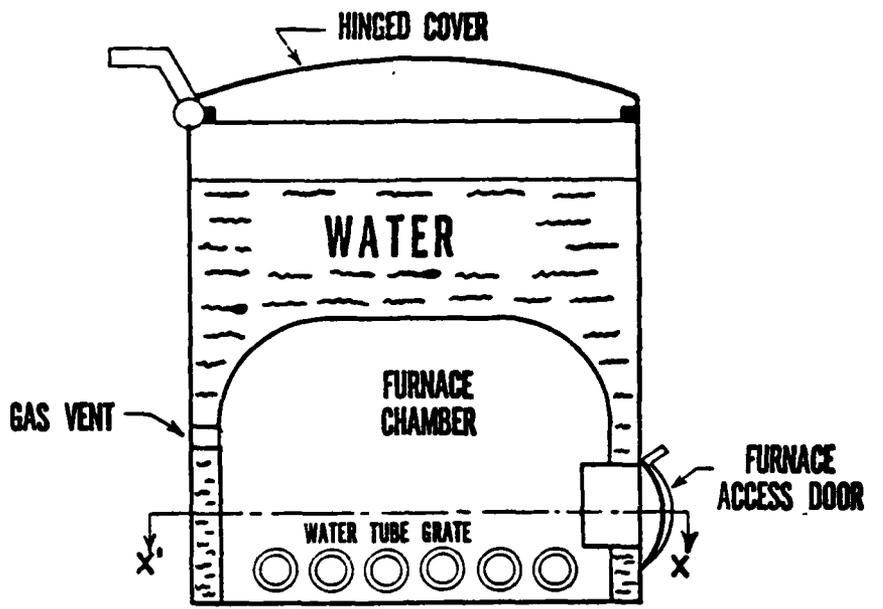
FIGURE 1: BRONZE WATER-COOLED TUBES AND WATER-COOLED FURNACE CHAMBER WALL

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It was not until the latter part of the 17th century that
boilers in a real sense appeared on the scene. During this
period, the greatest potential use for power seemed to be
pumping water out of mines, and a number of engines were in-
vented for this purpose.

These pumping engines, from those first successfully
pioneered by Thomas Savery in 1699, to the engine patented
in 1769 by James Watt, were not true steam engines, but atmos-
pheric engines. Steam was used to fill a cylinder in which a
piston was mounted. When the steam was suddenly condensed by
a spray of water, a partial vacuum was created, and the piston
was forced down by atmospheric pressure. The steam pressure
that raised the piston, with the help of a counterweight on
each cycle of operation, was only slightly above atmospheric
pressure. The steam pressure was so low that without the
counterweight the piston would not have moved. Consequently,
the boilers for these engines were of relatively simple design.

The wagon boiler, designed by James Watt, and in use
from about 1774, was hardly more than a long, riveted, metal
water tank placed over a brick setting or narrow brick oven.
The furnace was at one end and the stack of chimney at the
other. The combustion gases passed along the bottom of the
boiler to the other end where they were channeled back to
heat the sides before entering the chimney at the hearth end.
Although many odd forms were tried between 1780 and 1820, the
wagon boiler was most common until the invention of the

internal flue by Oliver Evans in the United States and, independently, between 1800-1805 by Richard Trevithick in England.

In true steam engines, where steam does mechanical work, it is advantageous to impart the highest pressure possible to the steam before it is released into the pressure cylinder. Both Evans and Trevithick saw the necessity of high pressure engines for marine and locomotive applications and were chiefly responsible for the adoption of higher steam pressures. Built in two forms, the boiler was a long cylindrical shell with a large internal cylindrical tube. In one form the flue contained a furnace at one end and opened to a stack at the other. In the second a brick setting was used. Hot gases from the furnaces were drawn through the setting, heating the lower surface of the shell, and then returned through the internal flue to the stack above the furnace.

With the addition of a number of small internal flues, called fire tubes, operating pressures increased rapidly. The usual pressure in the time of Watt was only 5-7 psi above atmospheric. Evans, Trevithick, and John Stevens of New York early in the 19th century used pressures from 50-75 psia. Jacob Perkins of Massachusetts, in work done between 1823 and 1827, actually obtained 1400 psia. But in general, fire-tube boilers are limited to relatively low operating pressures because of the difficulty of constructing a shell strong enough to withstand the high pressure required for efficient operation. As pressure demands of steam engines and turbines increased,

the fire-tube boiler was supplemented by the water-tube boiler.
During the 1820's, steam pressures on the Mississippi steam-
boats and elsewhere in the United States had been raised to
100 and 150 psia, but explosions were frequent.

It was early recognized that the strength requirements
for increased pressure could be met if the tubes contained the
water rather than the hot gases. Two problems impeded the
development of the water-tube boiler. First, adequate circu-
lation of the water within the tubes had to be maintained in
order to absorb the heat transferred through the tube wall.
Without this circulation the tube-wall temperature and the tube
would be burned out. A second difficulty was the explosion
hazard of high pressure operation where relatively large
amounts of water are in contact with heating surfaces liable
to fail because of overheating. In the event of such failure,
liquid water at the saturation temperature corresponding to
a high pressure, flashes to steam with explosive force. If,
however, the total amount of water in a boiler flows in a
sufficient number of parallel circuits, as in a sectioned
boiler, a failure in the heating surface will not be serious.
Although these problems were recognized by Joseph Eve and
Goldsworth Gurney in 1825 and 1826, the only successful water-
tube boiler prior to 1870 was that patented by Babcock and
Wilcox in 1867.

The period from 1870 to 1900 was one of improvements.
The multiple drum bent-tube (Stirling) boiler, successfully

introduced in 1893 is actually the most recent form of boiler construction. Subsequent improvements in materials and methods of construction increased the steaming capacity of the largest central power plant boilers from 30,000 lbm/hr in 1905 to 1,000,000 lbm/hr by 1935. In 1926, Iving Moultrop was responsible for the increase in operating pressures from 350 to 1200 psia. By 1935 this pressure had gone to 1600 psia. Modern plants are built to operate above critical temperature at 4500 to 5000 psia. The steam generator superheat temperatures have likewise increased from a maximum of 550°F in 1905 to 750°F in 1925 and 950°F in 1938. Modern plants are designed to operate at 1050°F to 1100°F.

   3. Boiler Types

   There are basically two types of boilers: fire-tube and water-tube boilers. All boilers that restrict the passage of the hot gases through flues or tubes where they transfer their heat to the surrounding liquid are called fire-tube boilers. In the water-tube boiler the flue gases pass over the exterior surface of the tubes while the water being heated passes within the tube. Another distinguishing feature between these two types of boilers is that the fire-tube boiler is supported by the setting sidewalls or the fire box and the water-tube boiler is usually suspended from the overhead steel work and columns.

a. Fire-Tube Boilers

Figures 2 to 10 show a historical development
of the fire-tube boiler from the first boiler used by Savery
in 1698 to the famous Cornish boiler designed by Trevithick.
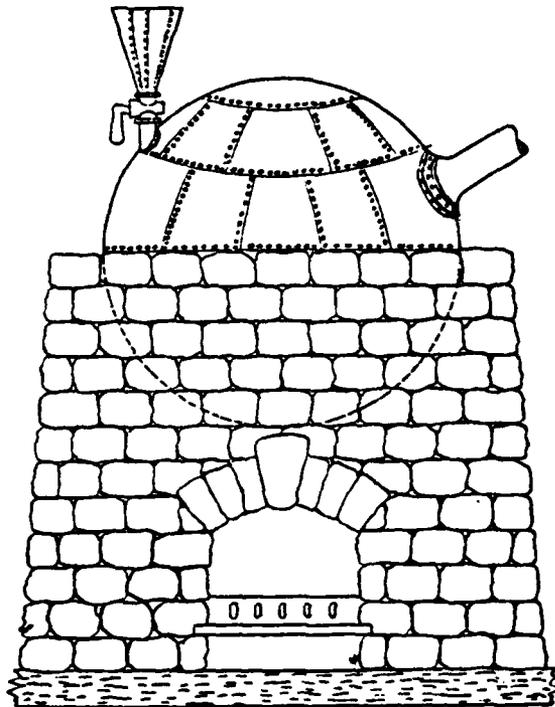
FIGURE 2: THOMAS SAVERY'S FIRST BOILER - 1698
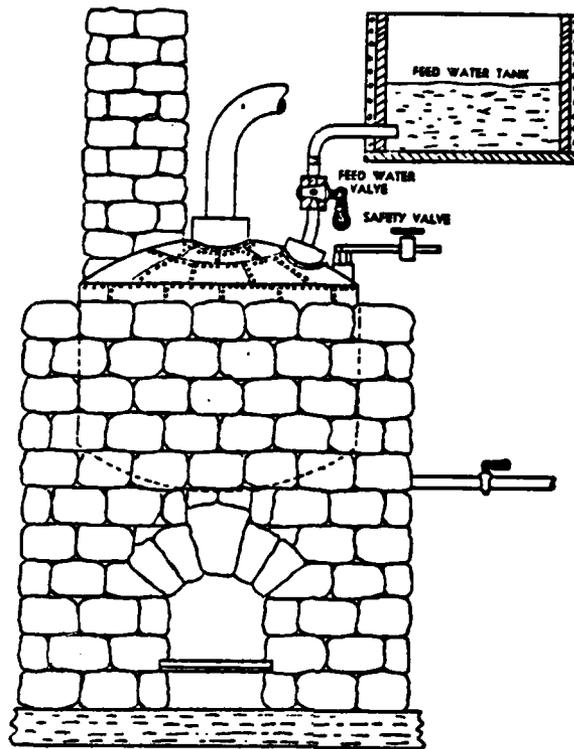
FIGURE 3: THOMAS SAVERY'S IMPROVED BOILER - 1702

Modern engineering practice limits the use of fire-tube boilers to about 500 horsepower and a working pressure of 150 psig. Designers of pressure vessels proportion the shell thickness to the diameter and working pressure to be encountered. Upon reaching the above limits the shell thickness requirement becomes such that it is difficult to fabricate. These limitations are also placed for safety reasons; for an explosion of one of the larger fire-tube
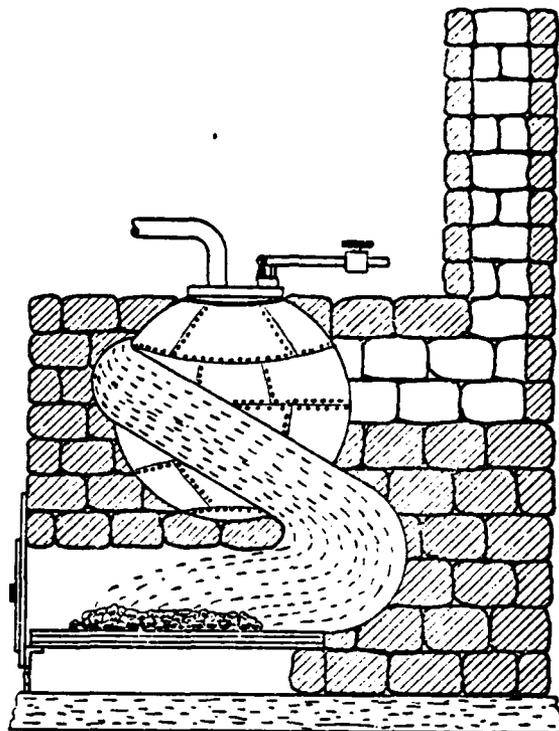
FIGURE 4:   DR. DESAGULIER'S WINDING FLUE BOILER - 1718

boilers can be quite disastrous due to the instant release
of large volumes of steam having tremendous expansive power.

       The Watt wagon-type boiler, Fig. 6, and the
cylindrical or flueless boiler, Fig. 7, are made up of flat
plates rolled to shape and riveted or welded longitudinally
and also circumferentially when several plate lengths are
required to obtain the desired capacity provided with a
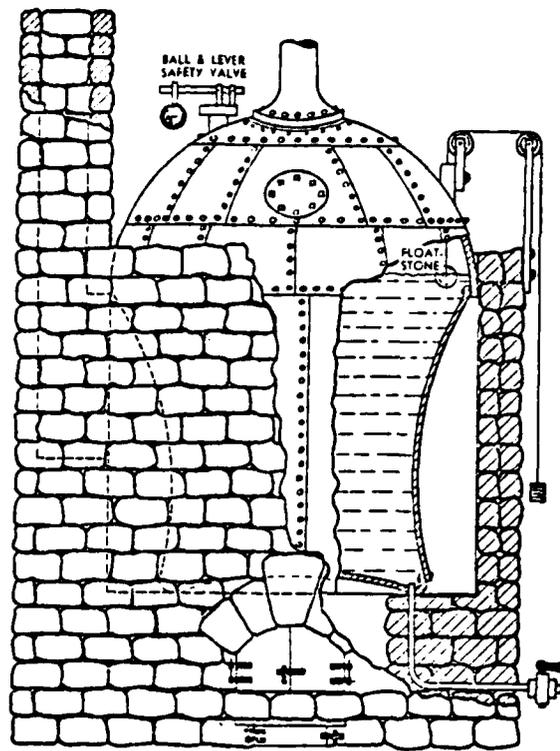
FIGURE 5: HAYSTACK BOILER - 1725
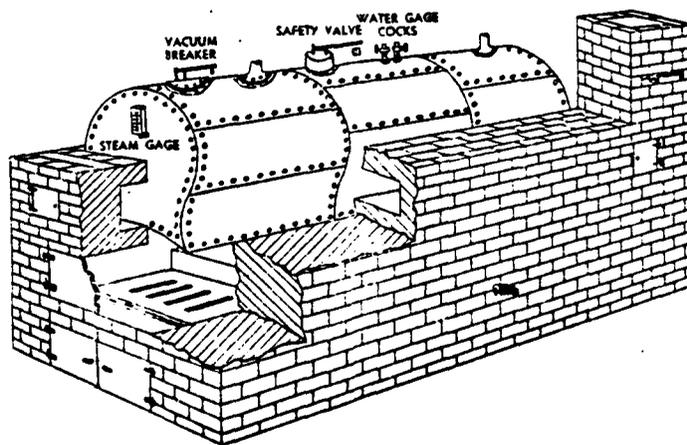
FIGURE 6: JAMES WATTS WAGON-TYPE BOILER - 1785

water-inlet, a condensate return, a steam outlet, a gauge
glass for ascertaining the water level, and a safety valve
for the relief of excess pressure makes the simplest kind of
boiler when mounted over a suitable refractory combustion
box or chamber and entirely insulated against heat loss.
This elementary design is rather awkward and requires much
space since the tank or drum is limited in its pressure with-
standing capabilities. The use of this type of boiler is
restricted to a moderate working pressure and stationary
applications only.
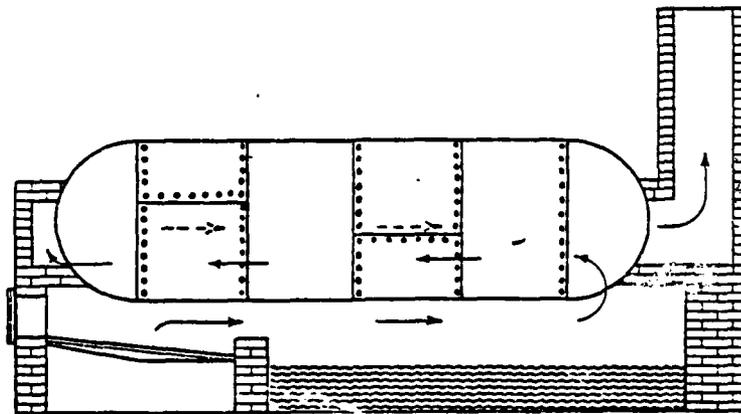
FIGURE 7: CYLINDRICAL BOILER - 1790

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Cylindrical boilers function by burning fuel in a refractory combustion chamber constructed under the cylindrical drum. The combustion products flow along the underside of the drum from front to back and then exit through an uptake to the chimney or stack. The heat in the flue gases must be transmitted through the metal drum into the liquid in order to preheat and vaporize it. Since the time element is short, with the gases passing from front to back only once, then exiting, the overall efficiency of this arrangement is low. The boiler is a notorious slow steamer; however, once the steaming action has started, it is a steady steamer. These boilers are limited to small capacities, usually to 350 boiler horsepower, because of their inherent tendency to explode.
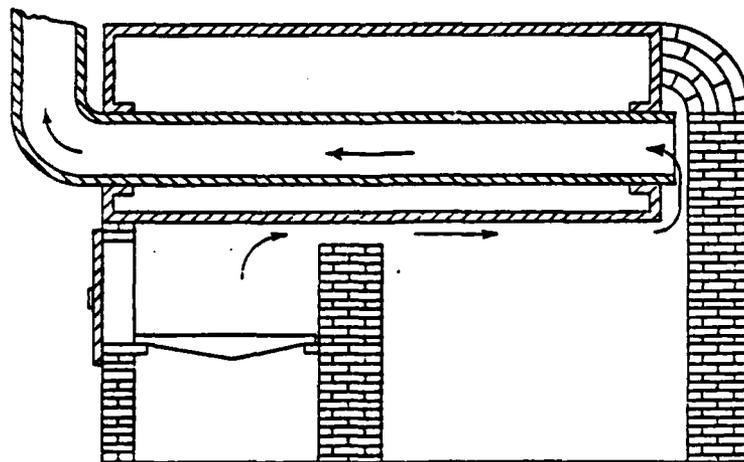
FIGURE 8: OLIVER EVAN'S RETURN-FLUE BOILER - 1800
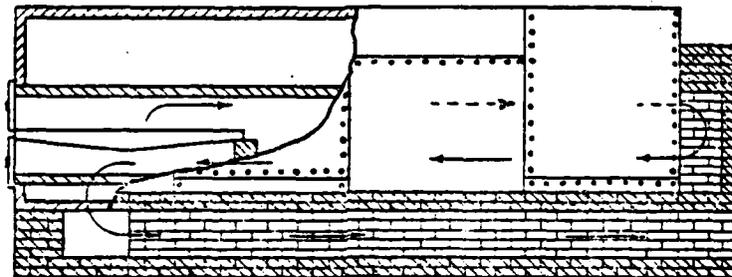
FIGURE 9:   RICHARD TREVITHICKS' CORNISH BOILER

The flue boiler is an improvement over the cylin-
drical boiler, see Figures 8 and 9, in that one or more inner
flues are incorporated for the purpose of conducting the flue
gases through the center of the shell, cylinder or drum.
There are many variations of design with some using these flues
as actual combustion chambers. The use of flues increases
the ratio of heating surface to the volume of water to be
evaporated; therefore, the flue boiler generates steam more
quickly than the ordinary cylindrical boiler. When the flue
boiler is arranged so that combustion takes place within the
flue greater savings in weight and space can be effected
rendering the design highly applicable to locomotion and
propulsion applications because of its portability.

The vertical upright fire-tube boiler, Figure 10, is avialibe in ranges from 5 to 300 horsepower, and consists of a cylindrical steel shell completely filled with tubes vertically mounted over a fire box. The flue gases pass in single flow directly through the tubes and into the stack at the top. While the vertical boiler is not highly efficient it has other virtues such as portability, a small floor requirement, a large heat surface in a small volumetric space, and an extreme ruggedness which has won it the name of "The Donkey Boiler". The popular fire engine "pumper" of several
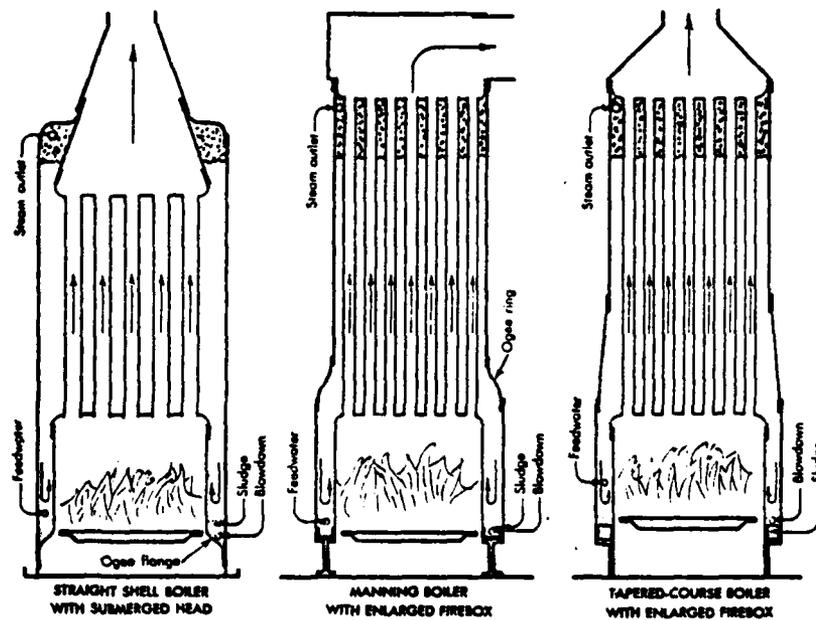
FIGURE 10: VERTICAL BOILER TYPES

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decades ago used this type of boiler to generate high pressure
steam in order to motivate a reciprocating piston plunger
pump. It also found wide use in the inaccessible, mountainous
terrain of the Pacific Northwest during the logging boom of
the late 1880's.

Fire-tube boilers become limited as capacity and
pressure requirements increase. Larger shell diameters re-
quire thicker plates to withstand the temperature and pres-
sure stresses. Temperature differentials in the boiler create
high stresses. These stresses combined with the effects of
precipitates and other deposits have caused many boiler
explosions [Ref. 2]. Other disadvantages include long warm-up
time requiring up to 15 hours to produce steam for propulsion,
unequal expansion of metal parts at an increased firing rate,
faster expanding fire tubes tend to pull loose from the boiler
shell, heavy weight, up to 15 lbs of boiler and water to pro-
duce 1 lb of saturated steam per hour and inefficiency
requiring large amounts of fuel to produce a relatively small
amount of steam.

Fire-tube boilers tend to be dangerous at high
steam pressures. This is partly due to inherent structural
problems and partly due to the relatively large quantity of
water and steam contained in the boiler. In the event of a
rupture in the boiler, the large amount of water and steam
releases great destructive energy. With this ever-increasing
demand for more and more steam at higher and higher pressure

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the water-tube boiler was developed. When steam and water at elevated pressure are confined within a tube of small or moderate diameter the thickness requirement is reasonable and practical.

b. Water-Tube Boilers

The water-tube boiler is composed of drums and tubes which are external to the drums and serve as interconnections for them. The drums store water and steam and contain no tubular heating surface, thereby allowing for these vessels to be much smaller in diameter than a fire-tube boiler shell and resulting in resisting higher pressures. This boiler may be a straight or bent-tube type, and in both cases, the tubes possess the entire heating surface.

The principle of using water tubes is an old one as shown by the use of the Pompeiian household boiler. Modern development of the horizontal, straight water-tube boiler began in 1766 by William Blakely who patented an improvement in the Savery engine which included a novel steam generator. The arrangement of heat transfer surfaces shown in Figure 11, probably was the first step toward the development of water-tube boilers, which has resulted in the modern high pressure water-tube boiler.

Three men, James Rumsey, John Fitch, and James Marlow, worked independently upon a box-shaped water-tube boiler with horizontal tubes, as shown in Fig. 12. The dis-advantages of this design were the weak box-shaped pressure
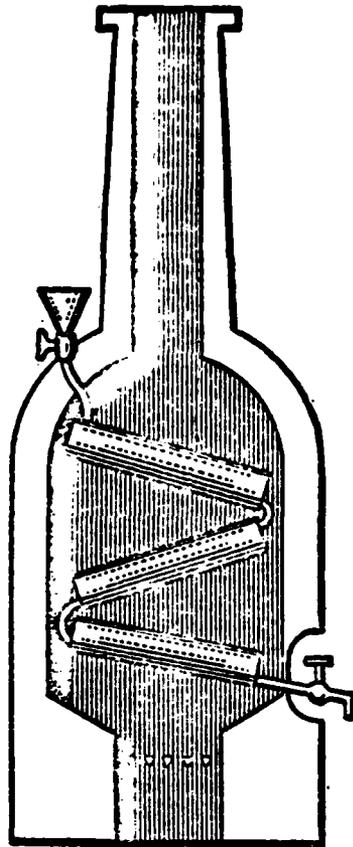
FIGURE 11: FIRST WATER-TUBE BOILER - 1766

vessels and the restricted water circulation in the horizon-
tal tubes. This was not critical since steam pressures were
between 3 to 7 psig.

In 1804, John Stevens designed a boiler for a
steamboat operating on the Hudson River. This boiler, Fig.
13, had slightly inclined tubes connected at one end with a

reservoir. The Steven's Porcupine boiler, as it is known, marked a definite advance in design of high pressure steam-generating apparatus, as its working pressure was 50 psig. Again, lack of proper water circulation in the tubes was a
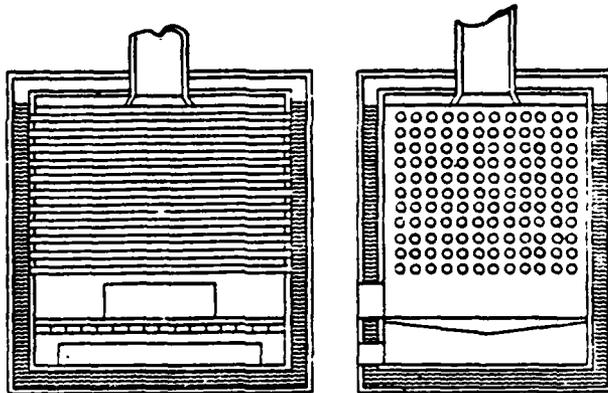
FIGURE 12: BOX-SHAPE WATER-TUBE BOILER - 1793
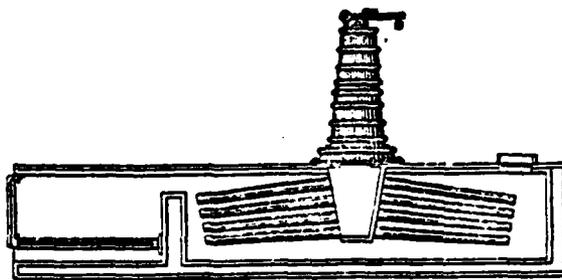
FIGURE 13: JOHN STEVENS' PORCUPINE BOILER - 1804

serious defect. Stevens' son, John Cox Stevens, designed a
water-tube boiler in 1805. This boiler, shown in Fig. 14,
consisted of 20 vertical tubes arranged in a circle, connecting
a water space at the bottom and a steam space at the top.
The steam and water chambers were annular spaces of small
cross section and small volume.

FIGURE 14: JOHN COX STEVENS' WATER-TUBE BOILER - 1805

Figures 15, 16, 17 and 18 show the development of
the water-tube boiler in the next sixty years. The first
sectional water-tube boiler with well defined circulation was
built by Joseph Eve in 1825. Sectional boilers have the

34
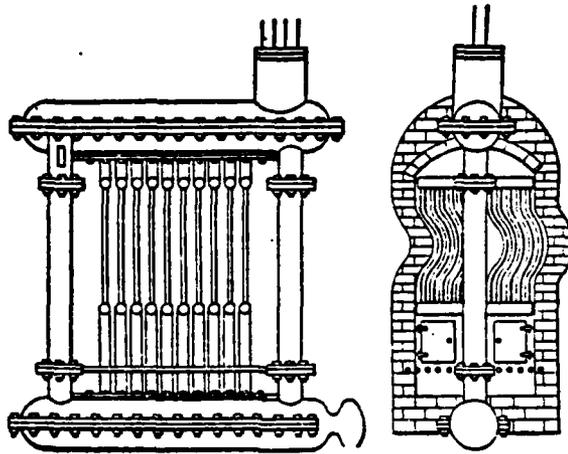
FIGURE 15:   THE FIRST SECTIONAL WATER-TUBE BOILER - JOSEPH
             EVE - 1825

water-steam spaces divided into small sections, none of which

is subject to disastrous explosion. Basically all water-tube

boilers are sectional in character. Figure 15 shows the

sectional composed of small tubes, practically vertical but

with a slight double curve. The tubes were fixed in larger

horizontal tubes called headers, which, in turn, were connected

to a steam space above and a water space below. The steam and

water spaces were joined by outside pipes (down-comers) to

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secure a circulation of water up through the sections and down
through the external pipes. In 1826, Goldworthy Gurney built
a number of boilers for use on his steam carriages, one of
which is shown below. This boiler consisted of a number of
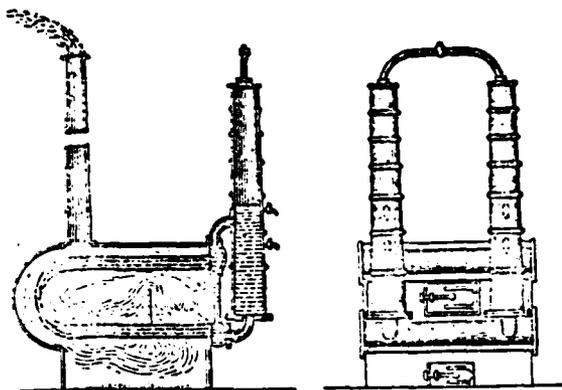
FIGURE 16: GOLDWORTHY GURNEY'S CARRIAGE BOILER - 1826

small "U" tubes, laid sidewise, with the ends connected to
larger horizontal pipes. These, in turn, were connected by
vertical pipes to permit circulation and also were connected
to vertical cylinders serving as steam and water reservoirs.
In 1856, Stephen Wilcox first used inclined water-tubes to
connect water spaces at the front and rear with a steam space
above, Fig. 17. The first to use inclined tubes in a sectional
form was George Twbill in 1865, Fig. 18. He used wrought iron
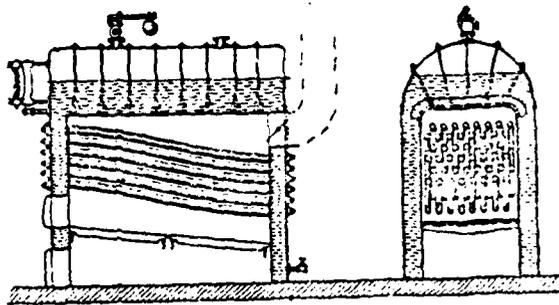
FIGURE 17: STEPHEN WILCOX'S BOILER WITH INCLINED TUBES - 1856
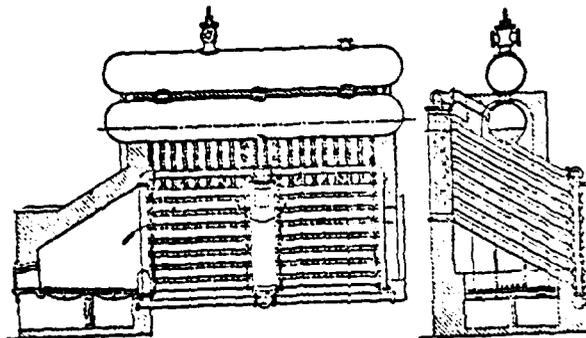
FIGURE 18: FIRST SECTIONAL WATER-TUBE BOILER WITH INCLINED TUBES - TWBILL - 1865

tubes connected at the front and rear to standpipes. These standpipes carried the steam to horizontal cross-drums at the top, the entrained water being separated and led to the rear standpipe. In the following years, development of water-tube boilers continued with great vigor. Steam drums soon were substituted for the nest of cast iron tubes and pipes origi- nally used for steam and water storage. Steel replaced cast iron for the generating tubes. Large tubes were replaced by small tubes to increase heat transfer areas. The transition to water-tube boiler design didn't arrive until the end of the 19th century when the triple expansion reciprocating steam engine was introduced to the navies of the world.

The head type boiler was the first form of water- tube boiler to displace the fire-tube type for use in steam vessels. A header type boiler was installed in the yacht Reverie in 1889 with a designed pressure of 225 psig. The success of this and other similar installations led to a decision by the Admiralty to use this boiler in the British Navy. In 1896, the U. S. Navy installed this type in three vessels, Marietta, Annapolis, and Chicago.

Although the Reverie design was an outstanding success and proved to be far more satisfactory in service than other boilers previously used by the Navy, combustion was not entirely satisfactory. The shape of the furnace made it necessary to slope the grates down toward the rear to gain furnace volume. At high ratings, this furnace was not suitable

for good combustion. Also, the fittings and water gages were not readily accessible from the firing aisle because of the steam drum location.

To overcome these difficulties, the design was modified to permit firing from the opposite end. Combustion conditions improved since the furnace was enlarged in the direction in which combustion took place. The grates were relocated and the accessibility of the tubes for cleaning from the fireroom floor was improved greatly. The steam drum was now at the firing end of the boiler, where the water gage could be seen easily and all fittings were more accessible. These modifications, seemingly very simple, revolutionized marine water-tube boiler design. The many improvements throughout the years have resulted in the sectional header type boiler, shown in Fig. 19, as we know it today.

The necessity for more steam brought on experimentation with forced draft to accelerate combustion of marine boilers. Although Robert Stevens tried in 1827 to supply air by forced draft, it was not until 1880 before a successful forced draft system was designed by James Howden. Interestingly, his design also included means by which waste heat in flue gases was used to heat the incoming air for combustion. The increased steam pressures and rapidity of steam generation led to serious trouble arising from overheating of tube ends and tube-sheets, causing many breakdowns and explosions from endeavors to work cylindrical fire-tube boilers under forced
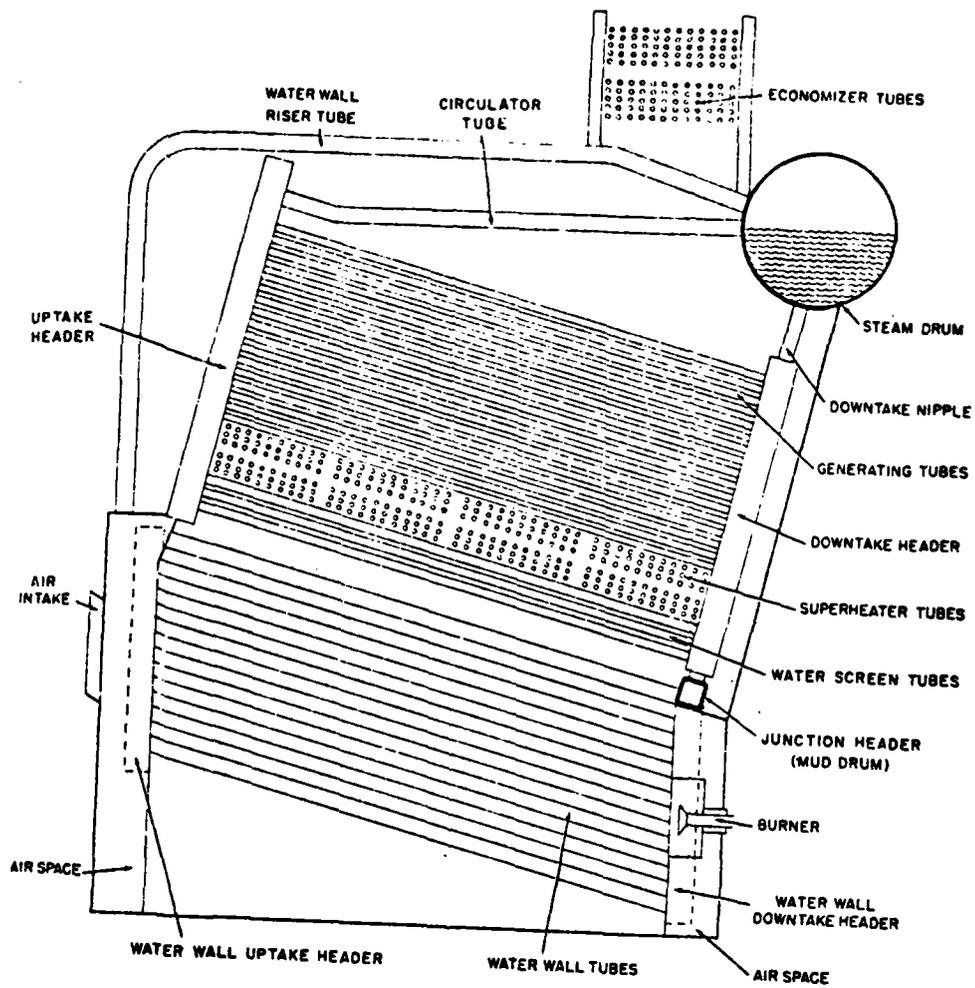
FIGURE 19: SIDE VIEW OF HEADER-TYPE BOILER

draft. The proponents of this boiler type, satisfied with the partial solution of installing tube inserts in the tube ends, argued that the generated pressures were sufficient for their needs.

Then in 1894, Charles A. Parsons developed and proved the superiority of the steam turbine for marine propulsion. The advantages of the turbine were immediately apparent and within a relatively short time (1894-1914) almost all important ships of the world navies were driven by turbines. Higher pressures, greater steam capacity and utilization of propulsion turbines concurrently accelerated the development of the water-tube boiler. A major contribution to higher pressure and greater steam capacity was the evolution of treating boiler water to prevent scale formation and boiler corrosion.

From the days of the caldron boiler until well into the turn of the 20th century there was very little improvement in steam generation except in the larger power-generating stations operated by the greater public utilities. At the end of the 19th century the new fast torpedo boats and destroyers, demanding boilers of light and compact design, brought forth the drum-type express boiler. The name "express" is used in its sense "dispatched with special speed" and is applied to this type boiler because of the "speed" with which steam can be raised and with which this boiler can answer steam demand changes.
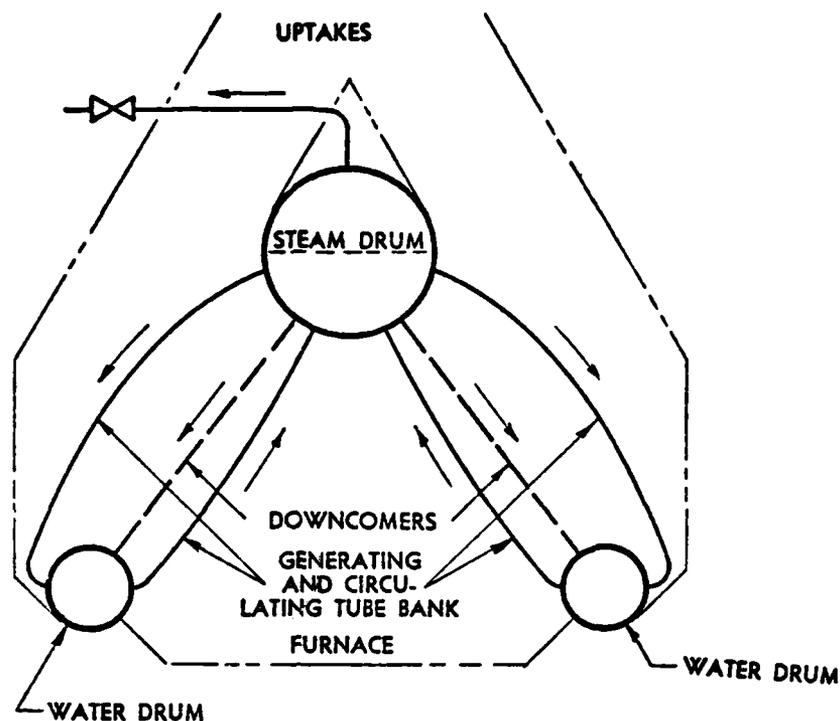
FIGURE 20:   BASIC "A" TYPE BOILER

The development of drum-type boilers as applicable to naval vessels to present time is shown in Figs. 20-29. Figure 20 illustrates the basic design of A-type boilers (so named because of the shape of the generating elements resembling the letter "A") which evolved from such early designs as the Yarrow express boiler shown in Fig. 21. The A-type boiler was originally designed as a coal-fired boiler (converted from coal to oil during the decade between 1905-1915) and was widely used for many years with only minor changes and additions.
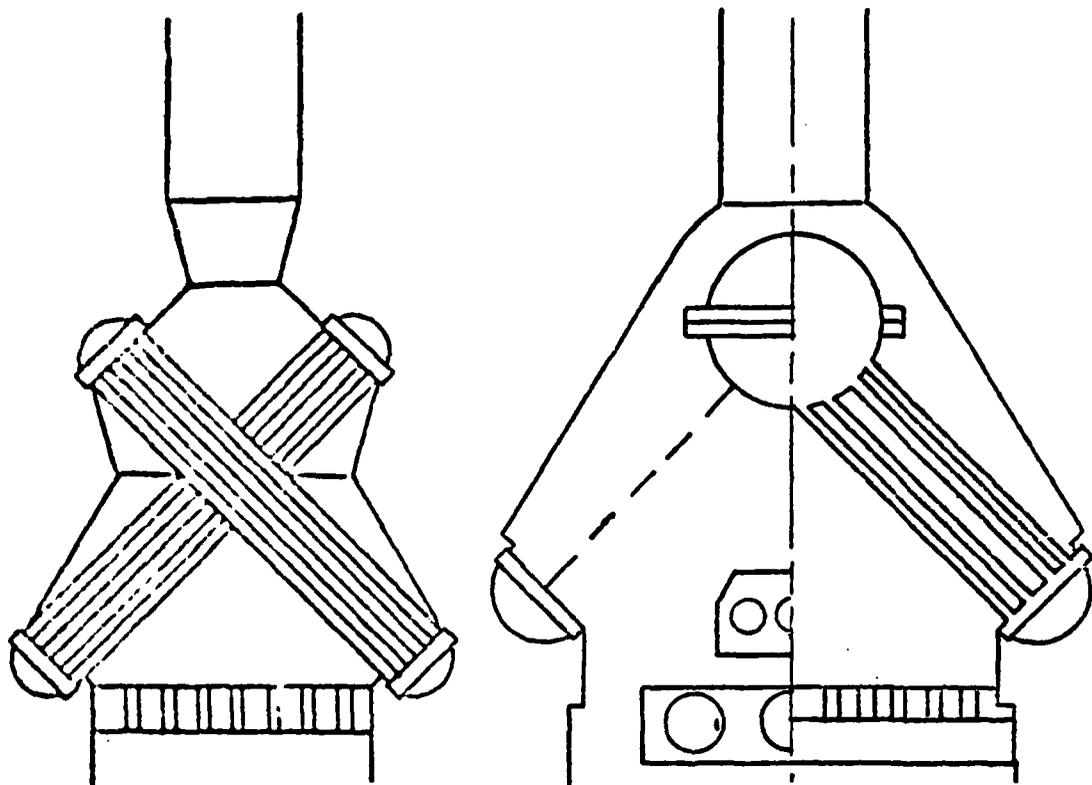
FIGURE 21: YARROW TYPE BOILER
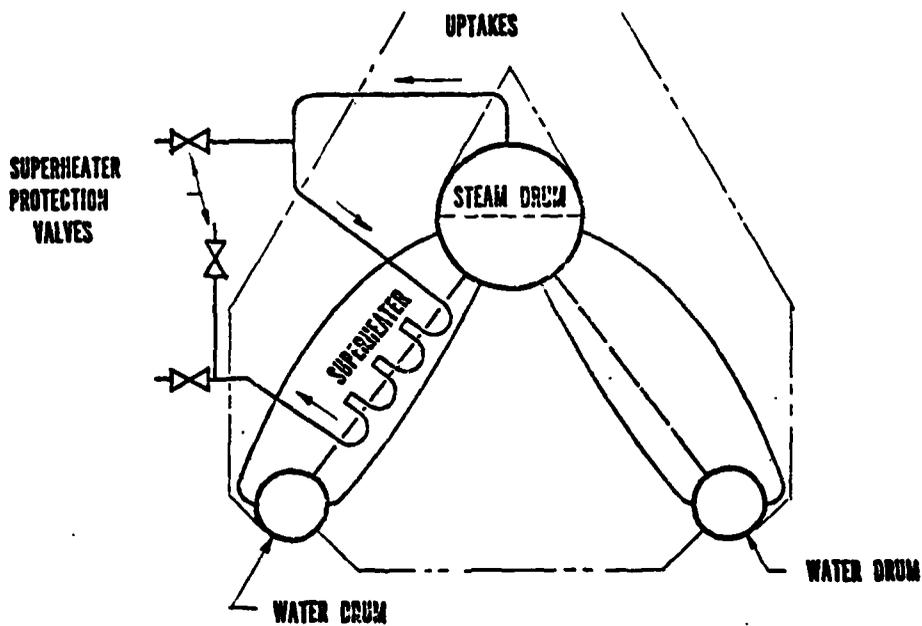
FIGURE 22: "A" TYPE BOILER WITH UNCONTROLLED INTERGRAL
SUPERHEAT

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FIGURE 23:   SECTIONAL EXPRESS TYPE BOILER - 1939

S.I.S. = SUPERHEATER INLET STOP VALVE
S.O.S. = SUPERHEATER OUTLET STOP VALVE
S.B.P. = SUPERHEATER BY-PASS VALVE
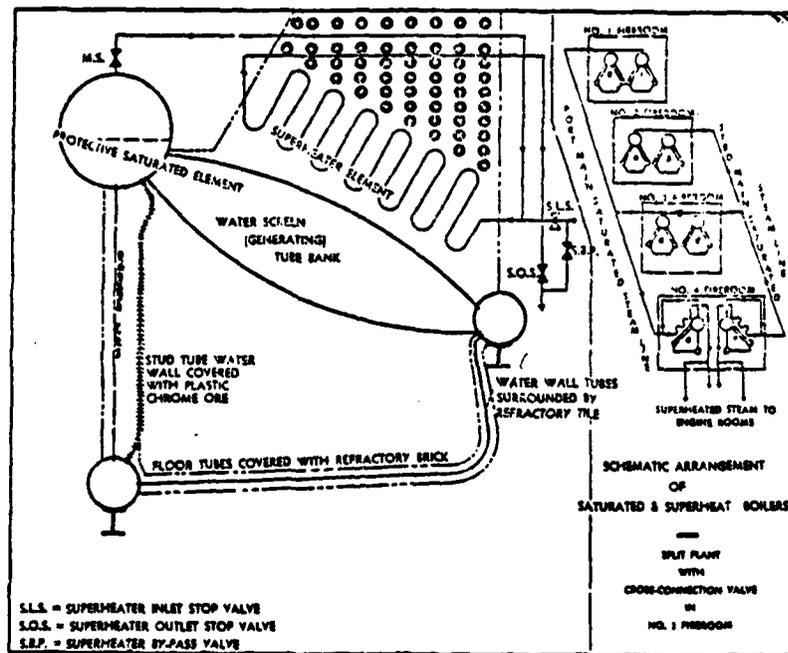
FIGURE 24:   SEPARATELY FIRED SUPERHEATER BOILER - 1930

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World War II created a huge demand for powerful,
packaged, compact steam generators simple to operate and easy
to maintain for use by the armed forces ashore and afloat.
Under this stimulus, designs were developed that are capable
of delivering two to three times the output of earlier boilers
containing the same heating surface. In some cases this can
be accomplished in one-quarter the previous volumetric space
requirement at great savings in weight and with efficiencies
running on the average of 80% [Ref. 3]. This was extremely
important to offset the increasingly congested space conditions
aboard destroyers.
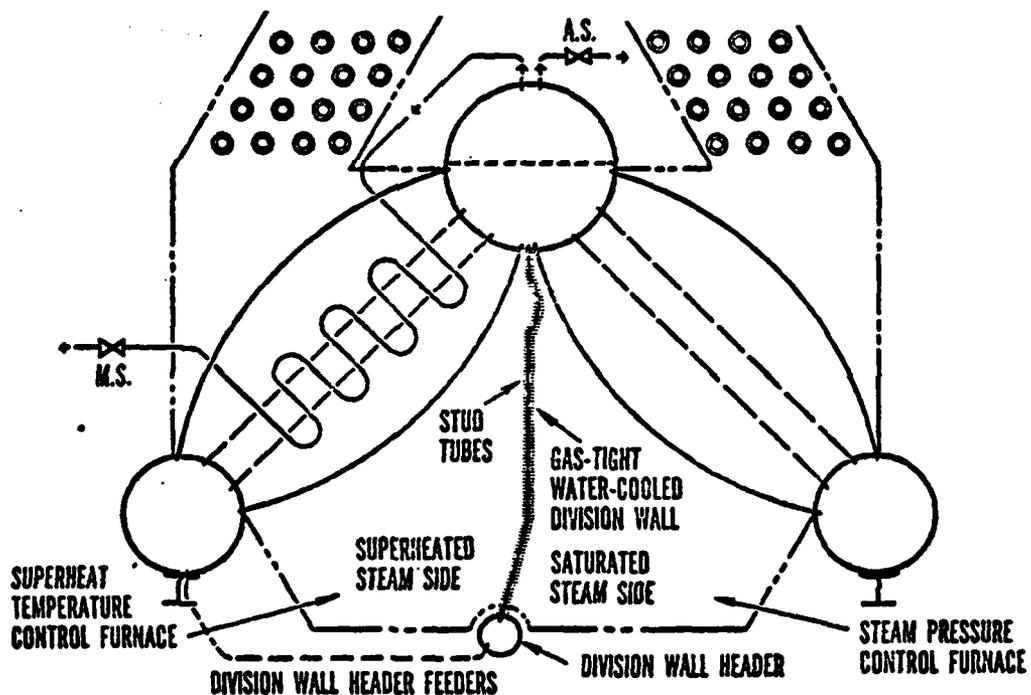
FIGURE 25: "A" TYPE BOILER FOR GENERATING SUPERHEATED STEAM
WITH CONTROLLED, INTEGRAL, INTERDECK SUPERHEAT

The general design most widely used for combat vessels built during WW II is shown in Fig. 26. The "M"-type boiler, as this design is commonly called because of the shape of its generating elements, was a two furnace, single-uptake, controlled superheat type boiler generally designed to operate at 600 psig-850°F. The modified "A" or Guest-type, Fig. 27, had a separately fired superheater of the radiant-convection type temperature control. However, this type superheater did not prove well adapted to naval requirements for rapid load changes and steaming at low rates for considerab. riod of
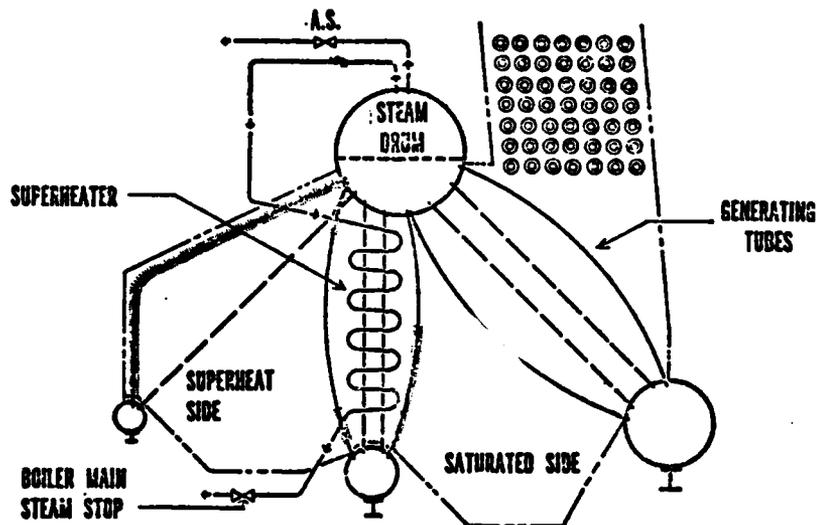
FIGURE 26: "M" TYPE BOILER WITH CONTROLLED, INTEGRAL SUPERHEATER
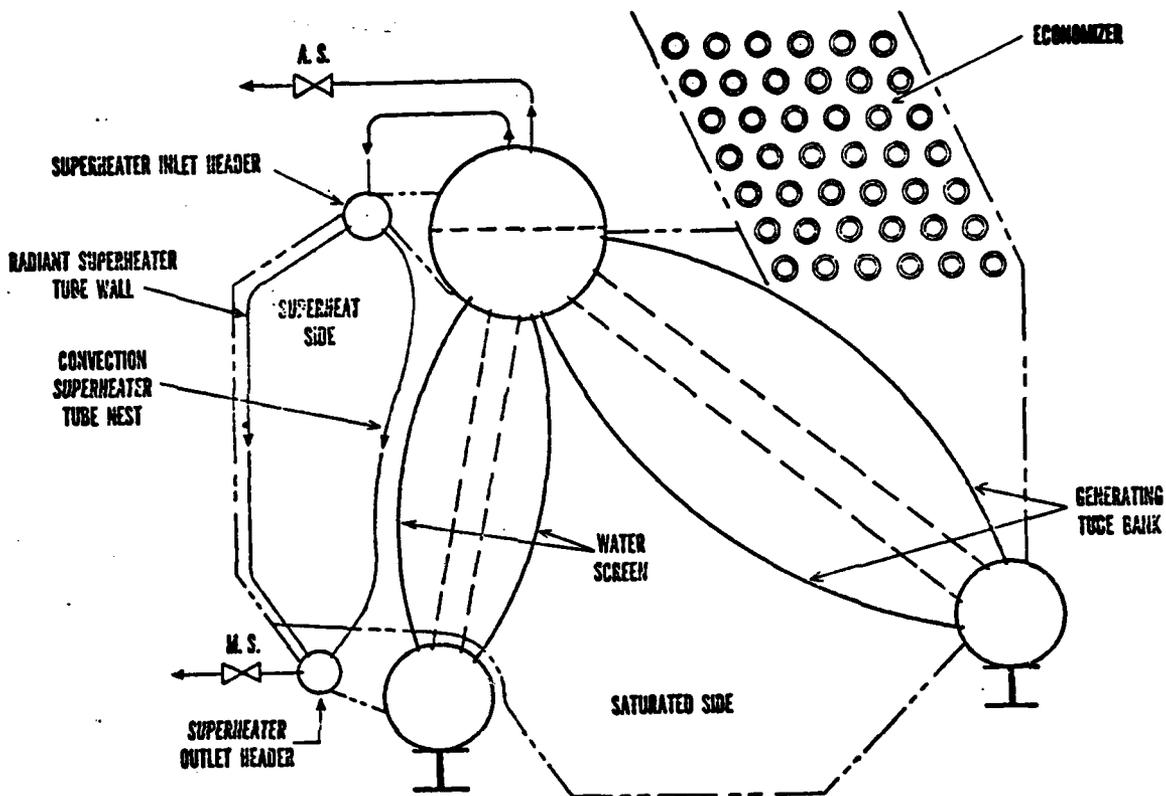
FIGURE 27: MODIFIED "A" OR GUEST BOILER

time. Another boiler used mainly in WW II destroyer escorts
is the single furnace boiler shown in Fig. 28. The "D" boiler
is the progenitor of boilers used in post-World War II ship
construction. Figure 29 shows the 1200 psig boiler which has
become the major naval fossil-fired steam generator.
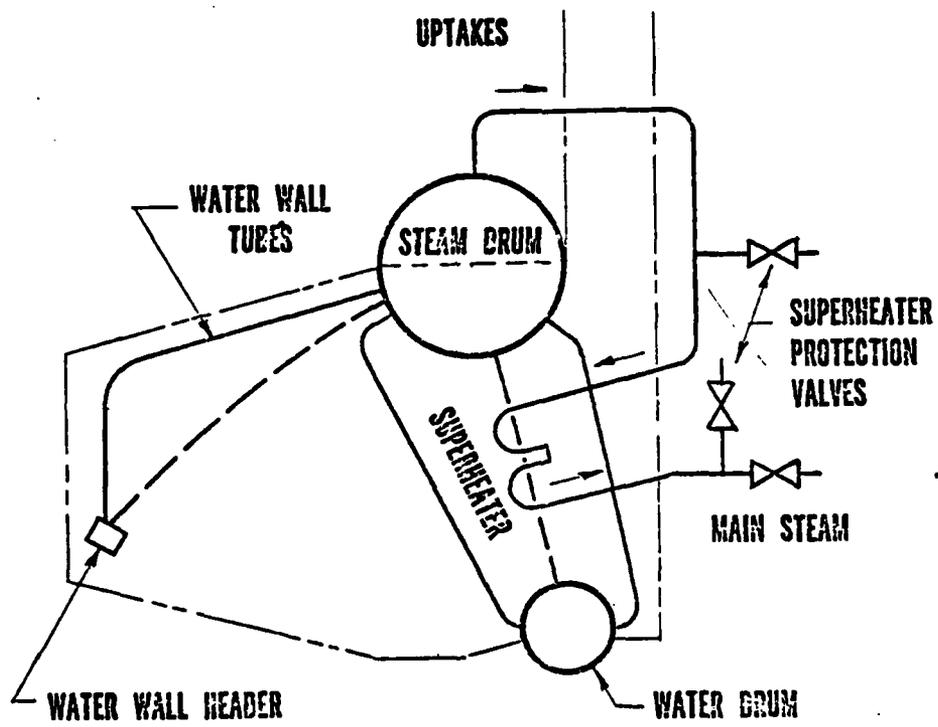
FIGURE 28: "D" TYPE BOILER
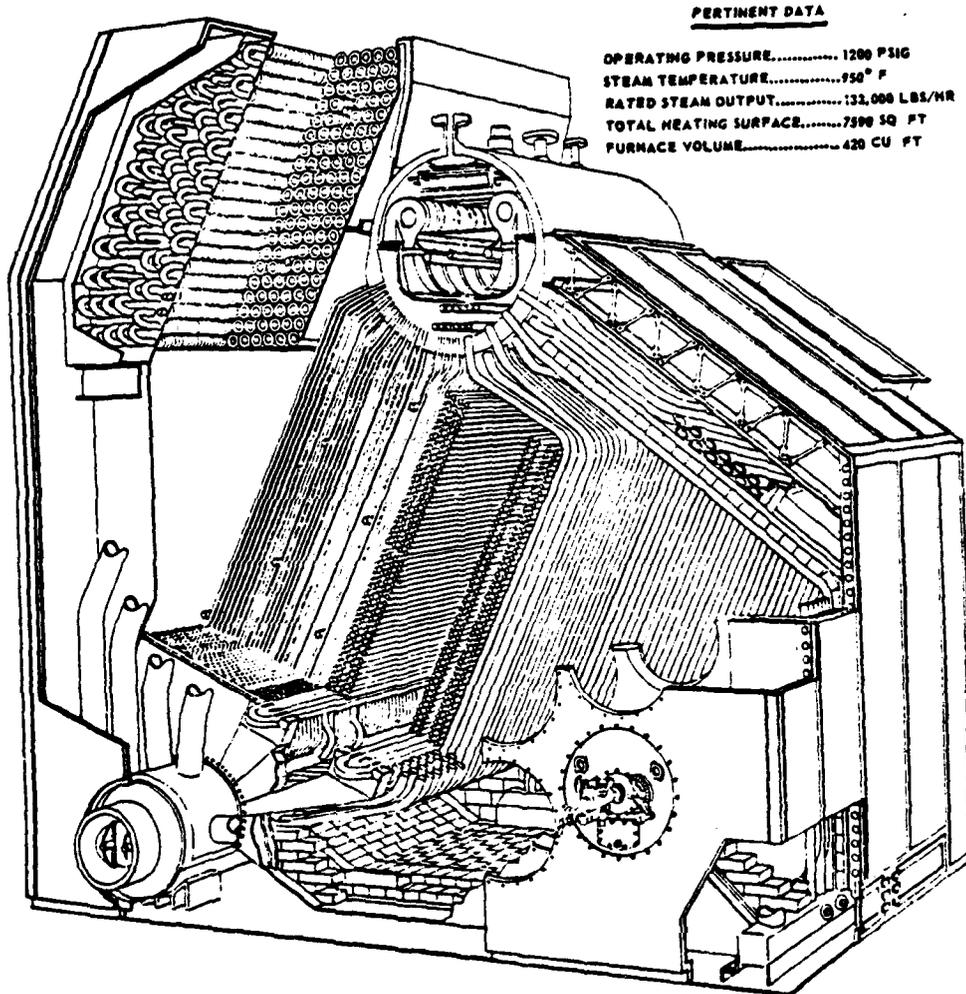
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FIGURE 29: NEWER 1200 PSIG SINGLE-FURNACE BOILER FOR POST
           WORLD WAR II DESTROYERS

4. Water Circulation

The majority of the previously discussed boilers have depended upon natural convection for the water-side circulation. That is, the difference in density between the water near the combustion chamber being heated and the cooler water in more remote parts of the boiler causes a natural flow away from the combustion chamber which in many cases is adequate. As boiler pressure increases, however, there is less difference between the densities of water and steam. At pressures over 1000 psig the density of steam differs so little from the density of water that natural circulation is harder to achieve than it is at lower pressures [Ref. 6]. At high pressures positive circulation boilers have a distinct advantage because their circulation is controlled by pumps and is independent of differences in density. The utilization of pumps to circulate and agitate boiler water increases heat transference, thereby increasing the output of a boiler without increasing the heating surface.

There are two main kinds of positive circulation boilers; one type is known as a controlled circulation or forced recirculation boiler and the other is known as a once-through or forced-flow boiler. In both types of boilers external pumps are used to force the water through the boiler circuits; the essential difference between the two lies in the amount of water supplied to the boiler.

In a once-through forced circulation boiler all (or very nearly all) of the feedwater pumped to the boiler is converted to steam the first time through without any recirculation. In its course along the full length of the tube, the water is heated, evaporated and superheated as additional enthalpy is applied. The steam generation rate is numerically equal to the water content supplied. Conventionally this type of boiler does not require a steam and water drum. Because no excess water is forced through the boiler, pumping other than that provided by feedwater pumps is not required. The very small amount of water that is not converted to steam in the generating circuit is separated from the steam and discharged to the feed pump suction. The once-through type of positive circulation boilers include the Benson and Besler boilers.
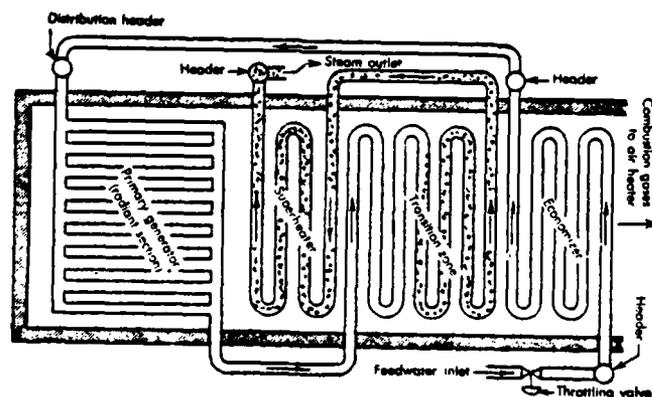
FIGURE 30:   THE "BENSON" BOILER

The Benson Boiler, Fig. 30, is characterized by the complete absence of steam separating drums. The unit responds sensitively to combustion rate changes and is therefore well adapted for use with a special turbine design in which variable pressure is used to accomodate a variable load. This boiler was first built in England (1923) by Mark Benson, a Czechoslovakian, but was developed by Siemens, Inc., in Germany. It has been built with capacities as large as 220,000 lbm/hr at 3200 psig.

Figure 31 shows the boiler circuit of a controlled circulation (or forced recirculation) boiler. In this boiler the water in the tubes is not evaporated to complete dryness but only to the point at which dissolved salts and solids are retained in solution. The mixture of water and steam passes to the steam-and-water drum where the steam is separated. The steam is passed through a separator where excess water is removed, then passed to a superheater. The separated water, along with feedwater, is returned to (or recirculated through) the heating circuit through the downcomers. The quantity of water passing through the boiler (circulation rate) is from 3 to 20 times the amount evaporated. This ratio requires constant speed recirculating pumps in addition to the boiler feedwater pumps.

5. Heat Source

The heat may be derived from (1) the combustion of fuel (solid, liquid, or gaseous); (2) the hot waste gases of
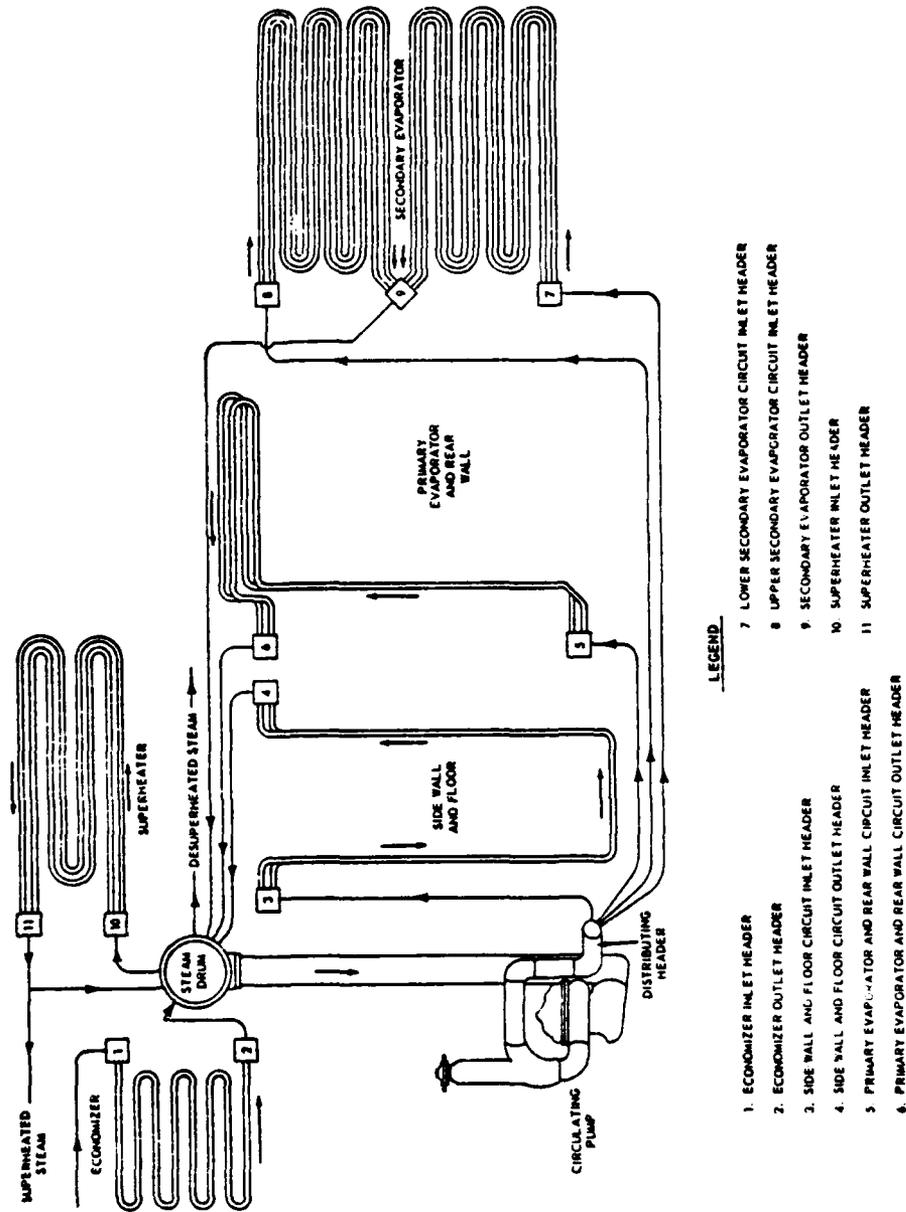
FIGURE 31: SCHEMATIC DIAGRAM OF CONTROLLED CIRCULATION BOILER

other chemical reactions; (3) the application of electrical energy; or (4) the utilization of nuclear energy.

Items 1, 3, and 4 are the most prevelant methods of supplying a heat source, but the increasing need for energy conservation has necessitated increased utilization of waste heat recovery.[1] Wherever a process waste product or gas is continuously discharged at a temperature of 1,000°F or higher, heat recovery should be considered. In addition to producing useful steam, the lowering of the flue gas temperature reduces maintenance of flues, fans, and stacks. One proposal for "by-product heat recovery" utilizes the hot exhaust gases from a shipboard gas-turbine propulsion plant as the heat source for a conventional Rankine cycle. This provides the means by which useful work can be extracted from a once wasted energy source.

B. RACER AND THE SST

The realization of finite petroleum resources has prompted researchers to accelerate the development of alternative fuels and energy sources, the employment energy conservation methods for short term relief, and the investigation of long term conservation techniques. The Navy, having made a major commitment to gas turbine propulsion systems, is currently investigating a means of fuel conservation without a reduction in steaming

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[1]Waste heat is a misnomer; a more appropriate term is "by-product heat."

time. The RAnkine Cycle Energy Recovery (RACER) system is being developed to recover some of the heat contained in the gas turbine exhaust gases.

The DD-963, FF6-7, and the DDG-47 class of ships are powered by the General Electric LM 2500 Marine gas turbine engine. The RACER system, an unfired waste heat recovery system, will function as a bottoming cycle for the LM 2500. Program requirements specify the development of a high efficiency cruise engine that utilizes hot exhaust gases to heat the working fluid (water) and convert it to useful energy through a steam turbine. It is intended that the design be compatible with the basic LM 2500 Marine gas turbine providing one shafts worth of power to the DD-963 and FFG-7, U.S. Navy surface cambatants. Further, specifications require provisions that will permit satisfactory inspection and cleaning of watersides/steamsides. It is important that internal surfaces can be properly observed for corrosion, scaling, defects, repairs, etc. Cleaning methods and techniques should be simple, easy to use, and practical for use with the proposed design.

The boiler, placed in the exhaust ducting, would by necessity have to be compact and lightweight, have a high steam generation rate and excellent response characteristics as well as simplicity in design, installation, and maintenance. LCDR Combs [Ref. 7], in his thesis conducted a feasibility study for propulsion assistance on waste heat recovery as applied to the Spruance class destroyers. Results of the study indicate that this

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proposal provides for substantial fuel savings based upon operational considerations. The proposed Rankine cycle included a once-through (O.T.) boiler having an optimal operating pressure of 600-800 psig.

For this application, the O.T. boiler is particularly attractive due to its small size. A special class of once-through boilers, which provides a possible design alternative, is the Short Straight Tube (S.S.T.) boiler. Also known as the "stagnation" (non-flow) boiler, it has excellent performance as a steam generation unit. This boiler has the same components as the O.T. boiler; however, in contrast, its construction consists of short (1-2 meters) and straight tubes with toroidal headers, Fig. 32. Due to boiler water level control and water chemistry problems, the S.S.T. boiler all but disappeared since the early 1900's. With the advancements made in water chemistry procedures and with the development of automatic boiler control systems, these problems should be alleviated, allowing present day application.

When compared to the normal forced or natural convection boilers, the once-through boiler is more desirable for waste heat recovery on the Spruance class destroyer. The O.T. boiler is lighter and safer, since no steam drum is required and the total mass of water is substantially less. However, a reveiw of its construction reveals, Fig. 33, a complicated arrangement of headers and long "serpentine" tubes. This leads one to question its applicability, especially in terms
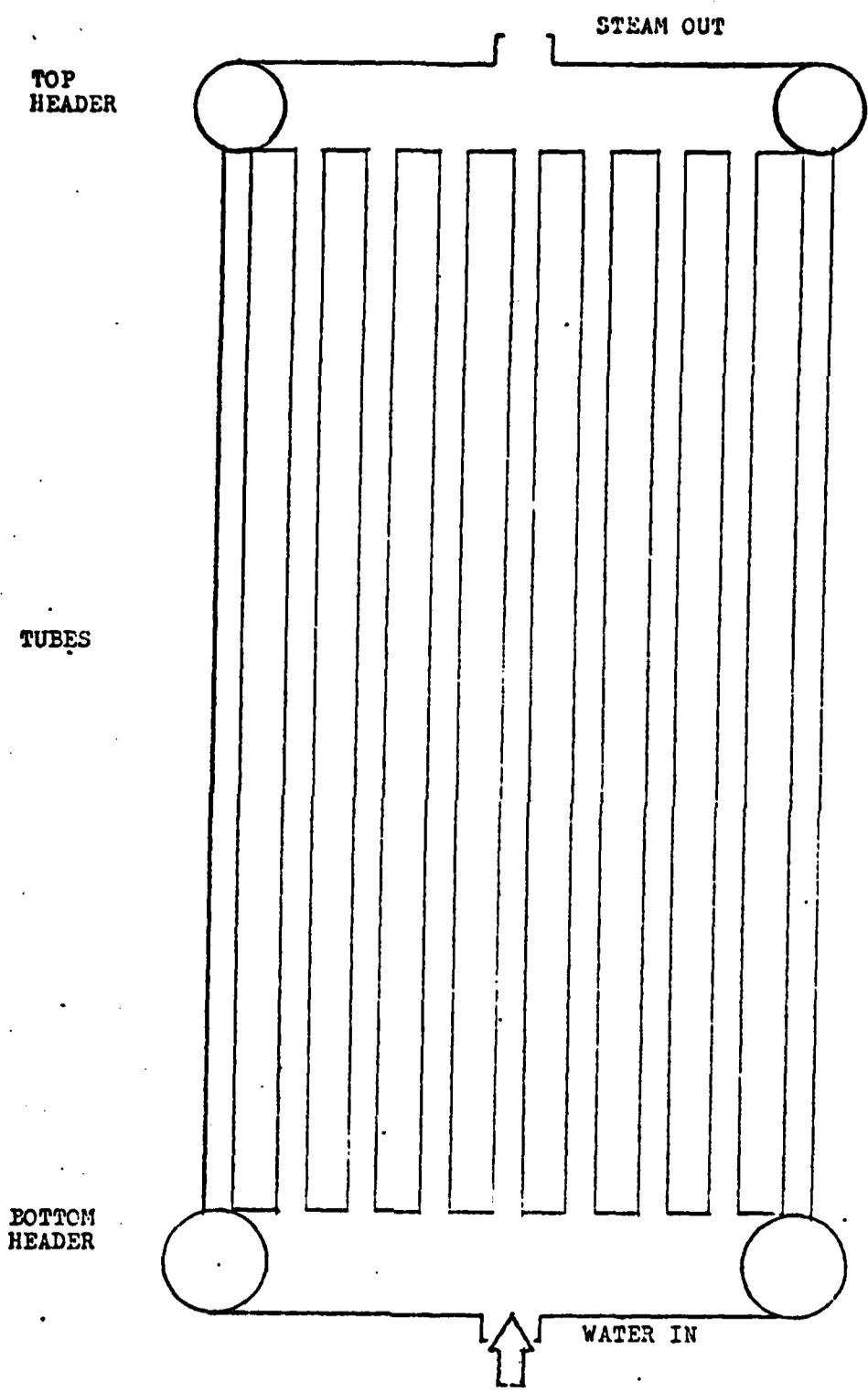
FIGURE 32:   SHORT STRAIGHT-TUBE BOILER UNIT

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of simplicity. In contrast, the S.S.T. boiler has a characteristic which makes it very attractive for this purpose. As it is the simplest of boiler designs, it lends itself to lower fabrication cost and simplified maintenance.

C. OBJECTIVE

The S.S.T. boiler is undergoing a resurgence in Japan for industrial usage due to its performance and construction [Ref. 8]. These industrial applications are for steam supplied up to 150 psig. In order to employ such a design at higher pressures, it is necessary to obtain some essential information.

In general, this need for information arises from three sources. First, the designer requires detailed information in order to provide for an optimal design. This entails a detailed quantitative study, for example, of heat transfer coefficients and two phase pressure drop. Secondly, operational conditions must be evaluated to ascertain the optimal steady state and transient operating conditions. Inclusive is casualty control data to diagnose faults due to the departure from the optimal conditions. Finally, and most paramount, is the requirement for the precise maximum safe operational limit. For these reasons, experimental investigation to ascertain design, operation and safety criteria is necessary. Firm establishment of the characteristics unique to the Short-Straight Tube Boiler is required to provide the means by which performance evaluation and comparison with other evaporators can be made. However, a lack of design and operational experience doesn't

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allow the engineer to anticipate S.S.T. characteristics a priori. Therefore, the logical beginning to experimentation is having a model from which predictions can be made.

The objective of this thesis therefore, is the development of a suitable model and computer program to provide the means for analytic study of the S.S.T. principle.

II. MODEL DESCRIPTION

A. AN OVERVIEW

The Short Straight Tube Boiler is modeled as a cross-flow heat exchanger with one fluid mixed; the other unmixed. The model is applied in a computer program for use in heat exchanger analysis for a phase change, or boiling situation. The same program could be easily adapted for general cross-flow heat exchanger performance study.

The two basic tools required to perform any heat transfer analysis are the first law of thermodynamics and the applicable rate equation.

A first law consideration of each fluid with mass flow rate $\dot{m}$, for an incremental area ΔA of the exchanger, yields

$$\Delta Q = (\dot{M}Cp\Delta T)gas = (\dot{M}Cp\Delta T)water \tag{1a}$$

Another expression for ΔQ is achieved using the rate equation and writing an expression for the conditions at ΔA; the resulting equation is

$$\Delta Q = U\Delta A \ (Tg-Tw) \tag{1b}$$

where Tg is the gas temperature in °F, Tw is the water temperature in °F, and U is the overall heat transfer coefficient in $BTU/(Hr \ FT^2 \ °F)$. For this analysis, U is assumed to remain constant throughout the incremental heat exchanger. Equations 1a and 1b can also be written as

$$\Delta Q = (\dot{M}Cp)_{min} \ \Delta T_{max} = U\Delta A(Tg-Tw) \tag{2}$$

which can be written as

$$\frac{\Delta T_{max}}{(Tg-Tw)} = \frac{U\Delta A}{(\dot{m}c_p)_{min}} \qquad (3)$$

For the incremental heat exchanger area, ΔA, the number of transfer units, ΔNTU is defined by the following equation:

$$\Delta NTU = \frac{\Delta T_{max}}{(Tg-Tw)} = \frac{U\Delta A}{(\dot{m}c_p)_{min}} \qquad (4)$$

The number of transfer units is a measure of a heat exchanger's ability to transfer heat. For a number of heat exchangers in series, the number of transfer units for the system can be written as follows:

let $C_{min} = (\dot{m}c_p)_{min}$

$$NTU = \frac{U_1\Delta A_1}{C_{min_1}} + \frac{U_2\nabla A_2}{C_{min_2}} + \ldots + \frac{U_n\Delta A_j}{C_{minj}}$$

or

$$NTU = \sum_{j=1}^{N} \frac{U_j\Delta A_j}{C_{minj}} \qquad (5)$$

where N represents the total number of incremental heat exchangers; ΔA_j's are the incremental heat transfer areas, and U_j's are the overall heat transfer coefficients for the $j\underline{th}$ elemental heat exchanger [Ref. 9].

For a particular type of heat exchanger, a mathematical relationship can be calculated between NTU, the number of transfer units; ε, the effectiveness of heat transfer; and C, the heat capacity rate ratio. If two of these quantities

are known, this mathematical relationship can be used to calculate the unknown. Reference [10] lists this relationship for cross-flow as follows:

$$C = \frac{C_{min}}{C_{max}} ; \qquad N = NTU = \frac{UA}{C_{min}}$$

C_{max} mixed, C_{min} unmixed

$$\varepsilon = (1/C) \left[1 - EXP \left\{ -C(1-e^{-N}) \right\} \right] \qquad (6a)$$

C_{max} unmixed, C_{min} mixed

$$\varepsilon = 1 - EXP \left[1(1/C \left\{ 1 - EXP(-NC) \right\} \right] \qquad (6b)$$

If the hotter fluid is a condensing vapor or the cooler fluid is a boiling liquid, the heat capacity ratio $C = 0$. For this case, Kays and London [Ref. 11] show that equation 6 reduce to

$$\varepsilon = 1 - EXP(-NTU) \qquad (7)$$

The proposed model utilizes this procedure to determine the overall heat transfer coefficient for the straight-vertical-tube cross-flow heat exchanger used as a boiler.

In the development of the model, the following assumptions were made:

1) A tube could be modeled as a group of N heat exchangers connected in series. The value of N is determined from fin geometry as the product of tube length times the number of fins per inch.

2) Each "elemental" heat exchanger has an overall heat transfer coefficient which can be considered constant over the segment area. With this assumption, equation 5 can now be applied to find the tube NTU.

3) The "elemental" applied heat flux (Q_s/A_{si}) is constant for each individual tube segment with Q_s given by

$$Q_s = \varepsilon (\dot{m}c_p)_{min}(T_g - T_w) \tag{8}$$

where Q_s is the segment heat transfer rate in BTU/hr, and ε is the segment effectiveness.

4) The water side pressure drop is negligible.

The SST model is divided into three principal sections: heating, boiling and superheating. The specified initial conditions for the model are: (1) gas temperature into the heat exchanger $(T_{g_{in}})$; (2) minimum average gas exit temperature from the heat exchanger $(T_{g_{out}})$; (3) gas flow rate $(\dot{m}_g)$; (4) water inlet temperature $(T_{w_{in}})$; (5) water/steam pressure (P_w); (6) amount of superheat at the tube exit (T_{sh}).

Since each tube is separated into N segments, it will be possible to determine axial temperature distribution of both the gas and water. At this point, a simplification should be noted; that is, what happens for any particular segment of any given tube is taken to be representative of the entire tube row. This varying gas-to-water temperature difference across each row of tubes is then utilized to determine a possible profile for the non-uniform gas temperature distribution across the entire heat exchanger.

B. GEOMETRY

Two different fin-tube configurations were selected for
this model. The fins can be either segmented or pin fins.
Tube banks can be arranged with either in-line or staggered
tube rows. From heat transfer and pressure drop measurements
made on in-line and staggered banks, Weierman [Ref. 12] stated
"it was seen that even tube rows in an in-line layout produced
about the same heat duty as four tube rows in a staggered
layout for the same pressure drop." He further stated "in-line
layouts should be reserved for those cases when a serious
justification exists. When cleaning lanes are required, a
staggered 45° layout should be considered." From this infor-
mation, the advantage of staggered layout is obvious, and it
is for these reasons that in-line banks are not considered.

The tube length, L_T, and the number of tubes per row, N_{tr}
are chosen by the designer. For this model the tube length
and heat exchanger height are constrained to that associated
with the S.S.T. boiler, that is, 3.2808 to 6.5616 feet (1-2
meters) in length. Additionally, it should be noted that the
maximum width is limited by shipboard space to 12 feet.

Since this model dissects a tube into N "elemental" heat
exchangers, all heat transfer areas are those associated with
one tube segment. These areas are essentially entire tube
areas divided by N; therefore, the following equations repre-
sent "elemental" heat exchanger areas.

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1. Segmented Fins

This fin profile is shown in Fig. 33 and the description of the segmented finned tubes is as follows:

d_i = tube inside diameter = 1.86 in.

d_o = tube outside diameter = 2.00 in.

N_f = fins per inch = 5.94

ℓ = fin height = 1.015 in.

ℓ_c = length of cut from fin tip = 0.82 in.

d_f = fin outside diameter = 4.03 in.

t_f = fin thickness = 0.048 in.

w_s = fin segment width = 0.17 in.

N_s = number of segments in 360 degrees = 38

In order to establish the minimum gas flow cross-sectional area the total "blocked" frontal area, A_b, of the "elemental" heat exchanger must be calculated from:

$$A_b = \frac{L_T d_o + 2L_T N_f \ell t_f}{N} \tag{9}$$

The fin surface area is

$$A_{fin} = [N_s(2\ell_c w_s + 2t_f \ell_c + w_s t_f) + \frac{\pi}{2}(\{d_f - 2\ell_c\}^2 - d_o^2)] \tag{10}$$

and the bare tube area is

$$A_{bt} = \frac{\pi d_o L_T}{N}(1 - t_f N_f) \tag{11}$$

2. Pin Fins

This fin profile is shown in Fig. 34 and the description of the pin finned tubes is as follows:

d_i = tube inside diameter - 1.86 in.
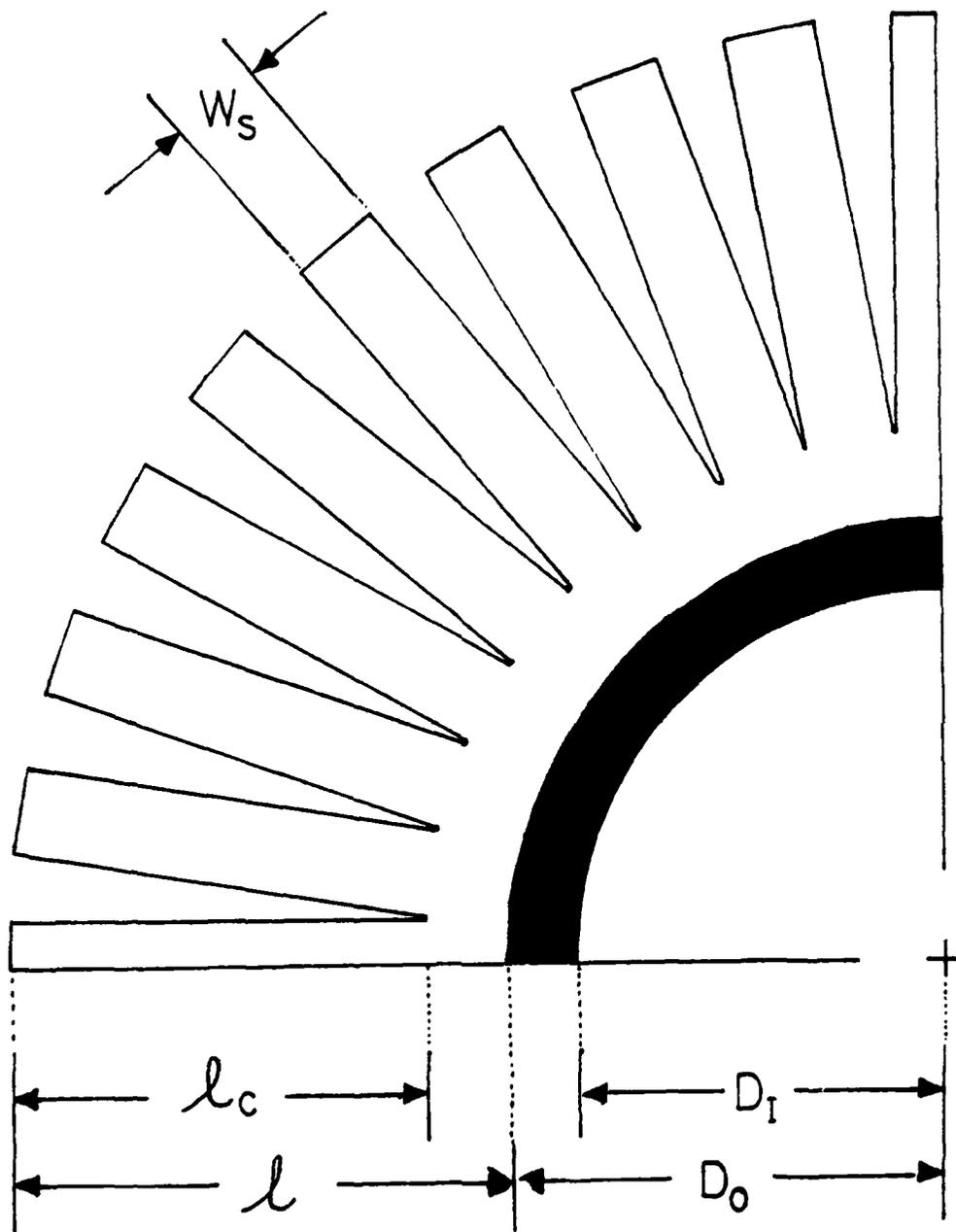
FIGURE 33: SEGMENTED FIN PROFILE
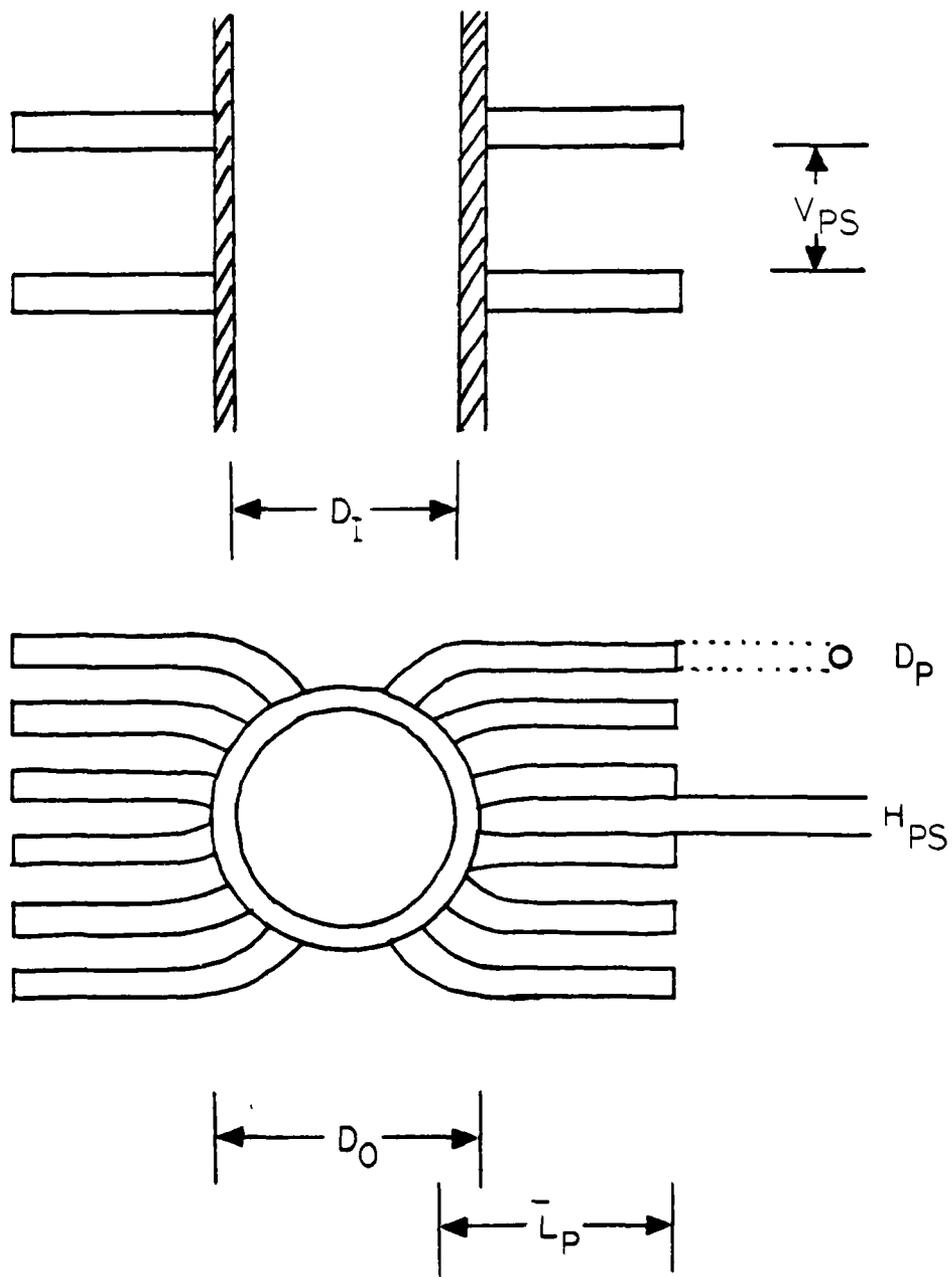
FIGURE 34: PIN FIN PROFILE

d_o = tube outside diameter = 2.00 in.

d_r = fin root diameter - 2.00 in.

N_f = pin groups per inch = 2.00

ℓ_p = average pin length = 1.015 in.

d_p = pin diameter = .125 in.

N_s = number of pins per group = 12

V_{ps} = vertical pin spacing = .375 in.

h_{ps} = horizontal pin spacing = .20 in.

The "blocked" frontal area is

$$A_b = \frac{L_T d_o + 2L_T N_f d_p 1_p}{N} \tag{12}$$

The fin surface area is found from

$$A_{fin} = \pi d_p N_s (1_p + {}^{d_p}/_4) \tag{13}$$

and the bare tube area is

$$a_{bt} = \frac{\pi d_o L_T}{N} (1 - N_s \frac{d_p^2}{4} N_f) \tag{14}$$

The previously determined areas are those which are particular to the given fin profile. With this information, the minimum gas flow area is

$$A_{min} = A_f - A_b \tag{15}$$

where A_f is the frontal area. The inside area available for heat transfer per segment heat exchanger is

$$A_{si} = \frac{\pi d_i L_T}{N} \tag{16}$$

and the outside heat transfer area is

$$A_{so} = A_{fin} + A_{bt} \qquad (17)$$

Finally, the cross-sectional fluid flow area is calculated from

$$A_{ff} = \frac{\pi}{4} d_i^2 \qquad (18)$$

For both segmented and pin fin-tube configurations, the center-to-center tube spacing in the transverse direction is 4.5 inches. Following Weierman's suggestion (45° staggered layout) leads to a spacing normal to the gas flow, S_n, of 4.5 inches and a spacing parallel to the gas flow, S_p, of 2.25 inches. This tube layout is shown in Fig. 35.
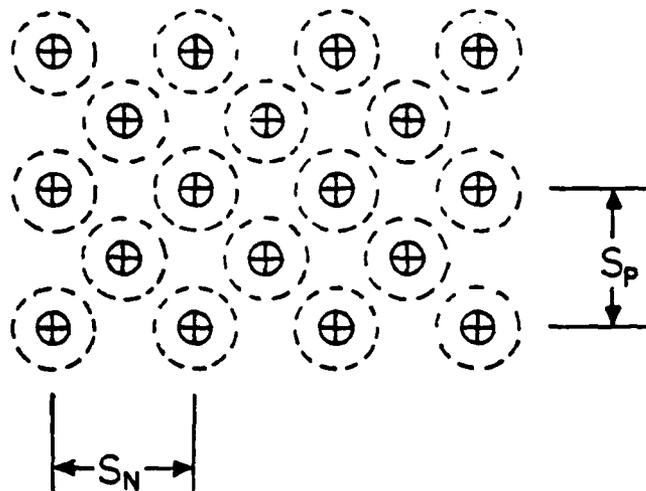
FIGURE 35: MODEL TUBE LAYOUT

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With the mass flow rates, terminal temperatures and heat exchanger geometry established, the remainder of the model may be solved for the number of tube rows or passes, actual iterim temperatures, and gas side pressure drop.

C. GAS-SIDE HEAT TRANSFER/PRESSURE DROP

For both the segmented and pinned-tube profiles, the gas-side Reynolds number is calculated initially using the gas bulk temperature to find the gas properties. With this Reynolds number, the segmented fin, j-factor, is obtained from a polynomial fit to the data for tube layout number 5 in Ref. [12]. The j-factor is related to the heat transfer coefficient h_g by the following relationship.

$$j = StPr^{2/3}$$

By introducing

$$St = \frac{Nu}{RePr}$$

the previous expression can be written as

$$j = \frac{Nu}{Re_g Pr_g} Pr^{2/3} = \frac{Nu}{Re_g Pr_g^{1/3}} \ ,$$

and

$$Nu = j \, Re_g Pr_g^{1/3} \ ,$$

where

$$Nu = \frac{h_g d_o}{K_g} \ .$$

Therefore, a relationship may be written for the heat transfer coefficient as follows:

$$h_g = j \frac{K_g}{d_o} Re_g Pr_g^{1/3} \ . \tag{19}$$

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For this equation all gas properties are evaluated at the film temperature.

For the pin fin, the gas-side heat transfer coefficient has contributions from the pins and bare tube, Equation 6-14 of Ref. [10], supplies the pin heat transfer coefficient

$$h_g = \frac{k_f}{d_p} C \, Re_g^N \, Pr_g^{1/3} \tag{20}$$

where the constants C and n are listed in Fig. 36.

Re_g	C	n
0.4-4	0.989	0.330
4-40	0.911	0.385
40-4000	0.683	0.466
4000-40,000	0.193	0.618
40,000-400,000	0.0266	0.805

FIGURE 36: HEAT TRANSFER CONSTANTS

Properties for use with the above equation are evaluated at the film temperature.

Eckert and Drake [Ref. 13] recommend the following relations for heat transfer from tubes in cross-flow:

$$Nu = (0.43 + 0.50 \, Re^{0.5}) Pr^{0.38} \left(\frac{Pr_f}{Pr_w}\right)^{0.25} \tag{21}$$

for $1 < Re < 10^3$

$$Nu = 0.25Re^{0.6} Pr^{0.38} \left(\frac{Pr_f}{Pr_w}\right)^{0.25} \tag{22}$$

$$\text{for } 10^3 < Re < 2 \times 10^5$$

For gases the Prandtl number ratio may be dropped, and fluid properties are evaluated at the film temperature. The presure drop for flow of gases over a bank of tubes may be calculated from

$$\Delta p = \frac{333.4 \; f'(G_{max})^2 N_T}{\rho_f} \left(\frac{\mu_w}{\mu_b}\right)^{.14} \tag{23}$$

Δp = pressure drop = inches H_2O

G_{max} = mass velocity = lbm/(FT^2-s)

ρ = free-stream density = lbm/FT^3

N_T = number of transverse rows

The emperical friction factor f' is given by Jakob [Ref. 14] as

$$f' = 0.25 + \left[\frac{0.118}{(S_n - d)/d^{1.08}}\right] Re_{max}^{-0.16} \tag{24a}$$

for staggered tube arrangements and

$$f' = 0.044 + \left\{\frac{0.08 S_p/d}{[(S_n - d)/d]^{0.43 + 1.13d/S_p}}\right\} Re_{max}^{-0.15} \tag{24b}$$

for in-line tube arrangements. For pin fins both friction factor expressions are required. Although the tubes are arranged in the staggered tube arrangements, the pins are essentially an in-line configuration.

The same equation is used for the segmented fin profile with the friction factor, f', obtained from a polynomial fit to the data for tube layout Number 5 in Ref. [12], (Fig. 5).

D. WATER-SIDE HEAT TRANSFER

Water-side heat transfer involved the heating, boiling and superheating of water or steam. A multitude of correlations are currently available to describe each individual mode. Although not prohibitive, a literature review resulted in several being chosen to represent the S.S.T. model.

With little known about this boiler type, four heat transfer correlations were candidates to represent the heating mode. One could reasonably surmise that the laminar situation is the most probable; turbulent correlations however, are also considered. An empirical correlation based on experimental data that takes into account the effect of varying physical properties and free convection is listed below [Ref. 15]:

$$h_{H_2O} = .17 \frac{k_f}{d_i} Re^{.33} Pr^{.43} \left(\frac{Pr_f}{Pr_w}\right)^{.25} \left(\frac{d_i^3 \rho^2 g_c \beta \Delta T}{\mu 2}\right)^{.1} \tag{25}$$

This relationship is valid for heating in vertical upflow for $Re < 2000$. Water properties are evaluated at the film temperature; Pr_w, at the wall temperature.

The work done by Sieder and Tate [Ref. 16], resulted in two simple empirical relationships which take into account fluid property variations. For laminar heat transfer in tubes, use the following equation

$$h_{H_2O} = 1.86 \frac{k}{d_i} (RePr)^{1/3} \left(\frac{d_i}{L_T}\right)^{1/3} \left(\frac{\mu}{\mu_w}\right)^{.14} \tag{26}$$

for

$$RePr \frac{d_i}{L_T} > 10$$

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and for turbulent heat transfer

$$h_{H_2O} = .027 \frac{k}{d_i} Re^{.8} Pr^{1/3} \left(\frac{\mu}{\mu_w}\right)^{.14} \tag{27}$$

for

.7<Pr<16.700

Re>10,000

For both situations, all fluid properties are evaluated at the mean bulk temperature of the fluid, except μ_w, which is evaluated at the wall temperature. Recommended by Dittus and Poelter [Ref. 17] for turbulent flow is

$$h_{H_2O} = .023 \frac{k}{d_i} Re^{.8} Pr^{.4} \tag{28}$$

The properties in this equation are evaluated at the fluid bulk temperature. This variety of expressions allows the designer to study a greater range of possibilities.

Consider now the conditions under which boiling will be initiated in the vertical heated tube. No boiling can occur while the temperature of the heating surface remains below the saturation temperature of the fluid at that particular location. It should be realized that fully developed subcooled boiling is not initiated whenever the heated surface first exceeds the saturation temperature. As shown in Fig. 37, a region of "partial boiling" exists between the subsaturation zone, A, and fully developed subcooled boiling, B. This "partial boiling" zone consists of comparatively few nucleation sites and in this zone, a proportion of the heat would be transferred by normal single-phase convection between patcher or bubbles.

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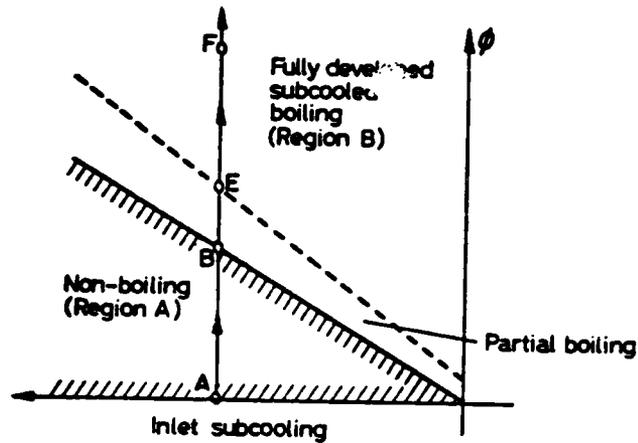
FIGURE 37: SURFACE HEAT FLUX VS. INLET SUBCOOLING

For forced convective flow, Bergles and Rohsenow [Ref. 18] developed a criterion based on analysis by Hsu [Ref. 19] and Han [Ref. 20] for the incipience of boiling. In this work, Bergles proposed a convenient numerical expression to determine the wall superheat required to initiate boiling. This expression is for the steam-water system in the 15-2000 psia pressure range.

$$\emptyset_{onb} = 15.60 \; p^{1.156} \; (T_{wall} - T_{sat})^{(2.3/p^{.234})} \tag{29}$$

75

where $\emptyset_{onb}$ (Btu/hr.ft^2) is the heat flux to cause nucleation at a wall superheat $T_{wall} - T_{sat}$ and at a system pressure p (psia).

As the surface temperature is increased further, the whole surface is covered by bubble sites, boiling is "fully developed" and the single-phase component reduces to zero. In the "fully developed" boiling region, velocity and subcooling have little or no effect on the surface temperature as observed experimentally. Throughout the partial boiling region Rohsenow [Ref. 21] suggests that

$$\emptyset_{Total} = \emptyset_{SPL} + \emptyset_{SCB}$$

where $\emptyset_{Total}$ is the total average surface heat flux, $\emptyset_{SPL}$ is the average surface heat flux transferred by single-phase convection, and $\emptyset_{SCB}$ is the average surface heat flux transferred by boiling. This method of superposition, shown in Fig. 38, utilizes the single-phase component given by

$$\emptyset_{SPL} = h_c (T_{wall} - T_{bulk})$$

where h_c is found by the familiar Dittus-Poelter equation.

The boiling contribution was successfully correlated with the experimental data using the equation suggested by Rohsenow [Ref. 22] for staturated nucleate pool boiling

$$\emptyset_{SCB} = \mu I_{fg} \left[\frac{g(\rho_f - \rho_g)}{g_c \, \sigma} \right]^{\frac{1}{2}} \left[\frac{c_p (T_{wall} - T_{sat})}{Pr \, C_{sf} \, I_{fg}} \right]^3 \tag{30}$$

where C_{sf} is a constant described by the liquid surface combination. The data reduced and correlation used a value of $C_{sf} = .006$. The surface-tension is calculated from [Ref. 23].
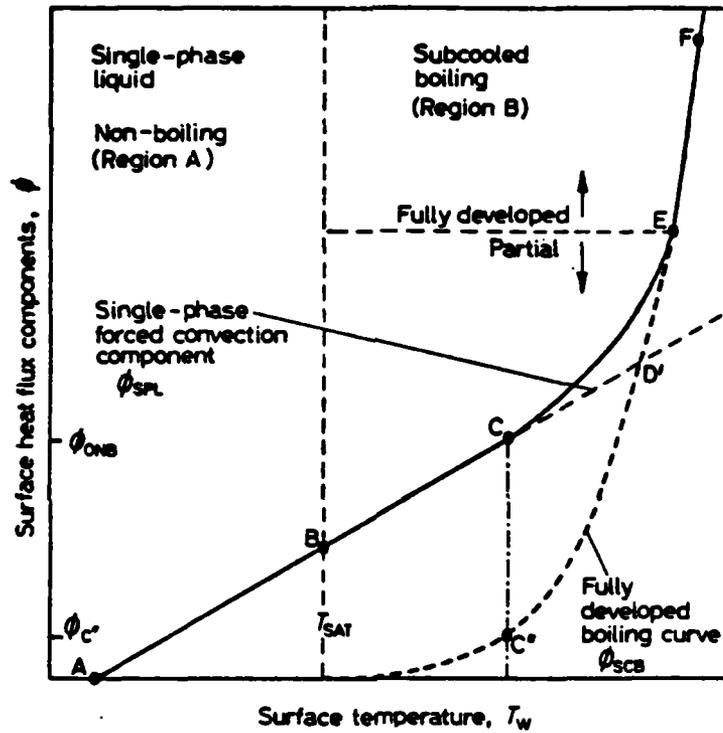
FIGURE 38:   METHOD OF ROHSENOW

$\sigma = 58 \times 10^{-4}$ (1 - .000142T) lbf/FT

with T in °F

In their introduction to Chapter 8, Hsu and Graham [Ref. 24] delineate the metamorphosis of two-phase flow typically shown in Fig. 39.

> A fluid enters the heating section as a subcooled liguid. Some distance downstream, subcooled boiling begins on the wall, but the bubbles do not leave the wall. Farther downstream, the bubbles begin to grow and depart into the main stream (low subcooled boiling region). As the fluid is near the saturation temperature, enough bubbles are generated to populate the whole cross section (bubbly flow) and finally coalesce into slug flow. However, slug flow exists only over a narrow range of valid fractions. As the void fraction is increased, the flow pattern quickly undergoes a transition into annular flow with a dispersed mist in the core (dispersed annular flow). The annular film is gradually depleted and thinned out. In the final stage, the flow is a mist flow with droplets carried by the superheated vapor. The wall region can be dry.

This excerpt from Hsu and Graham should raise the question of when do these individual changes occur. Reference [24] states that Haberstroh and Griffith found that the transition between slug and annular two-phase flow occurred at the void fraction between .8 to .9. The void fraction, α, can be determined from

$$\frac{\alpha}{1-\alpha} = \left(\frac{x}{1-x}\right) \left(\frac{u_f}{u_v}\right) \left(\frac{\rho_f}{\rho_v}\right)$$

with the vapor velocity found from

$$u_v = \frac{Gx}{\rho_v} \quad ; \quad G = \frac{\dot{m}_{TOTAL}}{A_{REA}}$$

and the fluid velocity is

$$u_f = \frac{G(1-x)}{\rho_f(1-\alpha)}$$

Dry-wall
mist

___

Annular
flow

___

Increased
quality
and
void

Slug
flow

___

Incipient
boiling

___
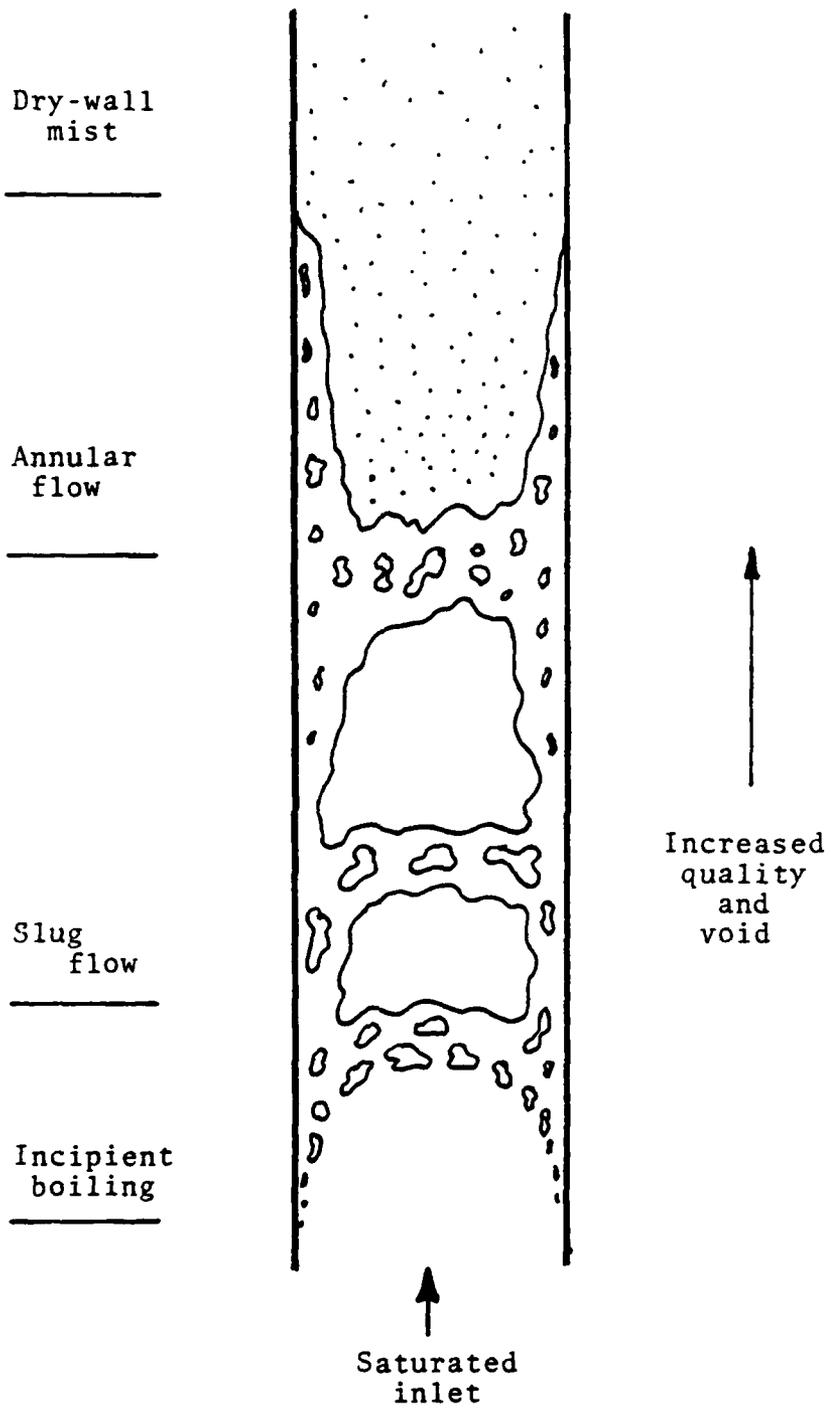
Saturated
inlet

FIGURE 39: TWO-PHASE FLOW DEVELOPMENT

where x is the mass quality, and is defined as

$$x = \frac{\dot{m}_v}{\dot{m}_{Total}} \qquad ; \qquad 1 - x = \frac{\dot{m}_f}{\dot{m}_{Total}} \qquad .$$

Now provided both liquid and vapor are in thermodynamic equi-
librium, i.e., they exist at the saturation pressure and tempera-
ture, then an alternative definition of mass quality x can be
given on the basis of thermodynamic properties.

$$x_e = \frac{I - I_f}{I_{fg}}$$

Only when the thermodynamic equilibrium exists are the values
of x and x_e identical.

However, this result was obtained for the adiabatic condi-
tion. Some important features of flow patterns in boiling
two-phase flow and differences between this and the adiabatic
case are discussed in Hsu and Graham. Furthermore, it has been
suggested that the Haberstroh-Griffith criteria for this
transition overpredict the flow quality. According to The
Handbook of Heat Transfer, [Ref. 25], Rohsenow observed that
the slug-to-annular flow transition occurs for qualities
between 5-10%. In this thesis a local true quality of 10% was
used as the accepted transition.

Slug flow occurs when the flow equilibrium enthalpy is
near saturation, the actual quality when $x_e = 0$ can be
determined by using the analysis for the subcooled boiling
regime. Based upon Hsu's [Ref. 19] postulate, Levy proposed
a criterion for bubble detachment, and therefore, a condition

for fully developed subcooled boiling. He showed that the
true quality can be related to the equilibrium quality through
the expression [Ref. 26]

$$x_T = x_e - x_d \ \text{EXP} \ (x_e/x_d - 1) \tag{31}$$

where

$$x_d = \frac{-c_p \ (T_{sat} - T_{bulk})}{I_{fg}}$$

Using Levy's approach, the fraction of heat flux used for
evaporation $\emptyset_{ev}$ can be determined from

$$\emptyset_{ev} = \emptyset_{Total} \left[1 - \text{EXP} \ (x_T/x_d - 1) \right] \tag{32}$$

At the present, a workable analysis of annular two-phase
heat transfer is not available; therefore, empirical correla-
tions must be employed. References 24, 25 and 27, contain
expressions derived from the work of Dengler and Addoms,
Schrock and Grossman, and Collier and Pulling, for example.
Chen [Ref. 28] compared the existing correlations and found
while the expression obtained by a particular research group
represented their own data fairly well, it failed to represent
any other groups' data with any reasonable accuracy. In
general, none of the existing correlations appeared satisfactory
for general use. As did Rohsenow, Chen postulated that two
basic mechanisms take part in the heat transfer process for
boiling of saturated fluids with flow.

The macroconvective mechanism, or single-phase convection,
is described by a modified form of the Dittus-Roelter relationship.

$$h_{mac} = .023 \frac{k}{d_i} Re^{.8} Pr^{.4} F \qquad (33)$$

In this expression the factor F is the ratio of the two-phase Reynolds number (Re_{TP}) to the liquid Reynolds number (Re), where Re_{TP} is defined by $Re_{TP} = ReF^{1.25}$. Chen reasoned and verified that this ratio is a function of the Martinelli parameter, X_{TT}, given by

$$X_{TT} = \left(\frac{1 - X_T}{X_T}\right)^{.9} \left(\frac{\rho_v}{\rho_f}\right)^{.5} \left(\frac{\mu_f}{\mu_v}\right)^{.1} \quad .$$

The work of Foster and Zuber was expanded to include the previously neglected effective and wall superheat difference in order to obtain an expression for the microconvective, or bubble nucleation and growth (boiling) term. A suppression factor given by

$$S = \frac{\Delta T_e}{\Delta T}^{.99}$$

where

$$\Delta T = T_{wall} - T_{sat}$$

ΔT_e = effective superheat with flow

was defined and later verified to be a function of $Re_{TP}(ReF^{1.25})$. Manipulating S through the Clausius and Clapeyron equation and combining with the modified Foster and Zuber expression, Chen obtained an expression for microconvection in terms of the suppresion factor and total superheat, ΔT.

$$h_{mic} = .00122 \frac{k^{.79} c_p^{.45} \rho_f^{.49} g_c^{.25}}{\sigma^{.5} u_f^{.29} I_{fg}^{.24} o_v^{.24}} \Delta T^{.24} \Delta P^{.75} S \qquad (34)$$

where σ is the surface tension of water previously defined, and

ΔP = pressure difference between the saturation pressure correspondint to T_{wall} minus the bulk saturation pressure and is in lbf/ft^2.

The total heat transfer coefficient is then obtained as the sum

$$h_{Total} = h_{mac} + h_{mic}$$

This correlation was tested with available data for water and organic fluids. The average deviation between calculated and measured boiling coefficients for all (over 600) data points from ten experimental cases was $\pm$ 12%. They include those for water in the pressure range of 1 to 35 atm with liquid flow velocities up to 14.7 ft/sec, heat flux up to 760,000 Btu/hr. ft^2, and quality up to 71%. However, no substantiating evidence was found to refute using this correlation up to a quality of 100%.

Computation of the water-side heat transfer coefficient from a true quality of 10-100% is accomplished by the utilization of the Chen correlation. Both the Reynolds number factor, F, and the Suppression factor, S, are obtained from polynomials fit to the data of Reference 27, Figs. 6 and 7 respectively. From the point of 100 percent quality to the tube exit, the evaluation of the water-side heat transfer coefficient becomes a choice of the previously discussed forced convection heat transfer correlations which are obviously dependent upon the Reynolds number.

E. OVER-ALL HEAT TRANSFER

Newton's law of cooling may be written as

$$Q = hA\Delta T \qquad .$$

It is often convenient to express the heat transfer rate for a combined conductive-convective problem in this form with h replaced by an overall heat transfer coefficient U.

The primary objective in the thermal design of heat exchangers is to determine the necessary surface area required to transfer heat at a given rate for given fluid temperatures and flow rates. The utilization of U facilitates this procedure in the above fundamental heat transfer relation

$$Q = UA\Delta T_{LM}$$

where ΔT_{LM} is the average effective temperature difference known as the log-mean temperature difference.

It is well known that the primary purpose of fins is to increase the effective heat transfer surface area exposed to a fluid in a heat exchanger. The performance of fins is often expressed in terms of the fin efficiency, n_f

$$n_f = \frac{[actual\ heat\ transfer]}{\begin{array}{c}heat\ transfer\ if\ entire\ fin\ were\\ at\ the\ base\ temperature\end{array}}$$

Analytical expressions for n_f are readily obtained for several common fin configurations.

For this model, both segmented and pin fins are considered to have ends that are at the free-stream temperature. Using a corrected length L_c (as was suggested by Harper and Brown) the insulated tip fin equation can be utilized, as follows

$$\eta_f = \frac{\tanh(mL_c)}{mL_c} \qquad (36)$$

where $L_c = L + {}^t/2$. However, for the fin dimensions utilized in this thesis $L \gg t$ which allows Equation 36 to be approximated by:

$$\eta_f = \frac{\tanh(mL)}{mL} \qquad (37)$$

where $m = \frac{h_g P}{k A}$

h_g = gas-side heat transfer coefficient

P = "wetted" fin perimeter

k = fin thermal conductivity

A = fin cross-sectional area

L = fin length

Now in terms of η_f, the heat transfer rate is given by the simple expression

$$Q = h (A_{bt} + \eta_f A_{fin}) \Delta T$$

remembering that A_{bt} is essentially

$$A_{bt} = A_{so} - A_{fin}$$

allows the heat transfer equation to be rewritten in terms of an "elemental" heat exchanger efficiency as

$$Q = h \eta_s A_{so} \Delta T$$

where $\eta_s = 1 - (1 - \eta_f) \frac{A_{fin}}{A_{so}}$

Using the tube wall resistance, together with the gas-side and water-side heat transfer coefficients and the "segment" efficiency, the overall heat transfer coefficient can be found as follows:

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(a) in terms of the inside area

$$Uo_i = \cfrac{1}{\cfrac{1}{h_{H_2O}} + \cfrac{A_{si}\ln(d_o/d_i)}{2\pi k_{tube}L_s} + \cfrac{A_{si}}{A_{so}\eta_s h_g}} \tag{38}$$

(b) in terms of the outside area

$$Uo_o = \cfrac{1}{\cfrac{A_{so}}{A_{si}h_{H_2O}} + \cfrac{A_{so}\ln(d_o/d_i)}{2\pi k_{tube}L_s} + \cfrac{1}{\eta_s h_g}} \tag{39}$$

Using the effectiveness - NTU method, an average "segment" effectiveness is calculated from equations 7a or 7b; if a phase change is occurring, equation 8. The expression for ε is now used in an iterative fashion. Each time an effectiveness is calculated it is compared to the previously found ε until a difference of less than .001 is obtained. This difference represents the convergence of the "elemental" heat exchanger effectiveness. During each iteration, the gas and water temperatures are determined in the following way. The total heat transfer rate is, per segment,

$$Q_s = \varepsilon C_{min} \Delta T_{max}$$

then the outlet gas temperature is

$$Tg_{out} = Tg_{in} - Q_s/C_{gas} \tag{40a}$$

and for the water/steam temperature

$$Tw_{ouy} = Tw_{in} + Q_s/C_{H_2O} \tag{40b}$$

or if boiling is present, the outlet enthalpy of water is found from

$$I_{out} = I_{in} + Q_s/\dot{m}_{H_2O} \tag{40c}$$

where   I = enthalpy in BTU/lbm

With these segment terminal temperatures, inside and outside tube wall temperatures may be calculated as follows:

$$Tg_{bulk} = \frac{Tg_{out} + Tg_{in}}{2} = Tg_b$$

$$Tw_{bulk} = \frac{Tw_{out} + Tw_{in}}{2} = Tw_b$$

Now performing a heat balance upon the "elemental" heat exchanger

$$Q_s = \frac{Tg_b - Tw_b}{R_{th}} \qquad (41)$$

with $R_{th} = \frac{1}{A_{si}Uo_i} = \frac{1}{A_{so}Uo_o}$

or $Q_s = \frac{Tg_b - T_{TO}}{R_o} = \frac{T_{Ti} - Tw_b}{R_i}$

here T_{TO} = outside tube wall temperature

T_{Ti} = inside tube wall temperature

$R_o = \frac{1}{n_s h_g A_{so}}$

$R_i = \frac{1}{A_{si} h_{H_2 0}}$

Combining to find the outside tube wall temperature in terms of the overall temperature difference as

$$T_{TO} = Tg_b - \frac{R_o}{\Sigma R_{th}} (Tg_b - Tw_b) \qquad (42)$$

and for the inside wall

$$T_{Ti} = Tw_b + \frac{R_i}{\Sigma R_{th}} (Tg_b - Tw_b) \qquad (43)$$

these wall temperatures are then utilized to find the gas-side and water-side film temperatures

$$Tg_f = \frac{Tg_b + T_{To}}{2}$$

$$\text{and } Tw_f = \frac{Tw_b + T_{Ti}}{2}$$

which are used in calculations requiring fluid property evaluation of the film temperature.

This completes the required calculations for the "elemental" heat exchanger. The final values of the water-side variables are used as the inlet conditions for the next segment, and the whole of the aforementioned procedure is repeated N (total number of elemental heat exchangers per tube) times. This represents the end of the tube.

At this point an overall tube heat balance is written based upon enthalpy considerations. The total heat transfer rate is found from

$$Q_{Total} = \sum_{j=i}^{N} Qs_j$$

$$\text{and } Q_{Total} = \dot{m}_{H_2O} (I_{exit} - I_{inlet})$$

solving for the water mass flow rate yields

$$\dot{m}_{H_2O} = \frac{Q_{Total}}{I_{exit} - I_{inlet}} \qquad \frac{lbm}{hr}$$

Obviously as the gas transverses the rows of tubes in the heat exchanger, its temperature decreases, producing a corresponding decrease in the gas-to-water temperature difference. The heat transfer rate being a function of this temperature difference also becomes less, however, the inlet and exit conditions are specified design parameters which remain constant. Since one of the assumed parameters was a water-side

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Reynolds number, this step allows for the actual mass flow
rate, and Reynolds number to be compared with the original.
If the relative error defined here as

$$E_r \equiv \frac{\dot{m}_{H_2O \; calc} - \dot{m}_{H_2O \; INITIAL}}{\dot{m}_{H_2O \; calc}}$$

is greater than 5%, the entire set of calculations is repeated
using the calculated mass flow rate as the initial $\dot{m}_{H_2O}$. In
this manner, a profile of the water mass flow rates is found
on a row basis for this type of heat exchanger.

A matrix of the segment outlet gas temperatures is utilized
at the inlet of the next row. The total number of rows is
obtained when the following condition is satisfied

$$Tg_o = \frac{1}{N} \sum_{j=i}^{N} Tg_{out_j} = \text{Design outlet gas temperature} \; .$$

III.   RESULTS AND CONCLUSIONS

A.   BACKGROUND

The foregoing model was applied in a computer simulation
program witten in BASIC for the Hewlett Packard 9845 model B
desktop computer. Inclusive are a set of supporting subpro-
grams for the determination of water, steam and air properties.
Finally, one subprogram was included to provide a choice for
the thermal conductivity of both the selected fin and tube
metal. With the establishment of the design variables to test
the model and to develop an understanding of the behavior of
the S.S.T., the program was activated to produce a heat ex-
changer design.

A review of the initial results revealed that the proposed
model required modification in order to produce more realistic
simulated data. The problem arose from the attempt to match
Rohsenow's and Chens correlation at 10% steam quality. It is
well known that boiling heat transfer is associated with rather
high heat transfer coefficients, and these results reflected
this fact up to the 10% quality point. The heat transfer co-
efficients were of the order of 20,000 BTU/HR FT^2 °F. However,
upon program transfer to the Chen correlation for the
determination of the water-side heat transfer coefficient, these
values were of the order of 6-20 BTU/HR FT^2 °F. Obviously,
this tremendous difference in magnitudes is definitely not a

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real simulation of the probable actual situation. This mis-match in heat transfer coefficients no doubt arises from the suppression factor, to utilize in the Chen microconvective term (Eq. 35), way below the lower limit of approximately 18,000 for the Reynolds number as shown in Figure 40.
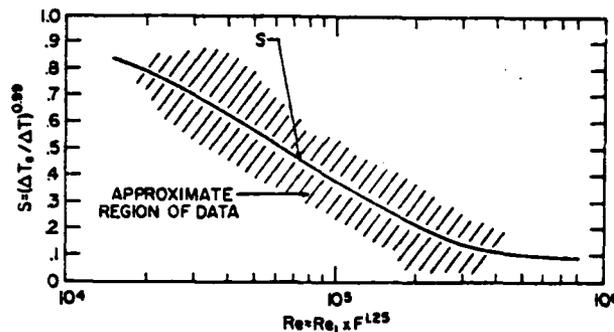
FIGURE 40: SUPPRESSION FACTOR, S

Therefore, the proposed model was changed to utilize the Rohsenow method for the entire range of 0-100% steam quality. The single phase contribution was calculated by using a value for the heat transfer coefficient, h_c, being determined by the local Reynolds number. Remember that initially h_c was cal-culated from the Dittus-Boelter correlation. A listing of the final program appears in Appendix A.

B. DESIGN VARIABLES

The following set of design variables was selected to produce the basic set of data.

1. S.S.T. Operating Pressure

The previous work of Combs [Ref. 7] indicated that the optimal operating pressure ranged from 600-800 psig. For this simulation, 600 psig was selected.

2. Frontal Diminsions

Space constraints for a DD-963- type engineroom limits the maximum heat exchanger width to 12 feet. To be consistent with this constraint, the frontal dimension of 3.658 m x 1m (12'x3.2808') was selected for the design.

3. Gas Flow Rate and Temperatures

The outlet from the gas turbine was taken to be 145 lbm/sec (522,000 lbm/hr) at 900°F under steady state and steady flow conditions.

4. Superheater Outlet Steam Temperature

The target superheat at the outlet was specified to be 50°F. At 600 psig this corresponds to 536°F. This amount of superheat will vary with water-side mass flow rate.

5. Water Inlet Temperature

The water inlet temperature was fixed at 50°F subcooling or 436°F.

6. Fin Geometry

The segmented fin profile was selected for the gas-side heat transfer.

C. THE DATA SET

A set of data was produced for the combination of design variables just discussed. This data was generated by varying the inlet Reynolds number, or if one prefers, the inlet mass flow rate, and was the result from the investigation of 60 different such numbers. The following set of output variables was considered.

1. Gas bulk temperature variation

2. Gas-side heat transfer coefficient

3. Overall heat transfer coefficient based upon the outside area

4. Tube wall temperature variation

5. Water/steam mass flow rate

6. Water-side heat transfer coefficient

7. Overall heat transfer coefficient based upon the inside area

8. Inside surface heat flux

9. Water-bulk temperature variation

10. Thermodynamic quality variation

This investigation included a Reynolds number range of 75 to 1650. Figure 41, shows graphically the water/steam mass flow rate corresponding to each Reynolds number in order to provide information concerning the range of mass flow rates involved. It should be re-emphasized that these Reynolds numbers are those associated with the water properties at the tube inlet temperature of 436°F. The outlet Reynolds number

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WATER MASS FLOW VS REYNOLDS NO.

FIGURE 41

94
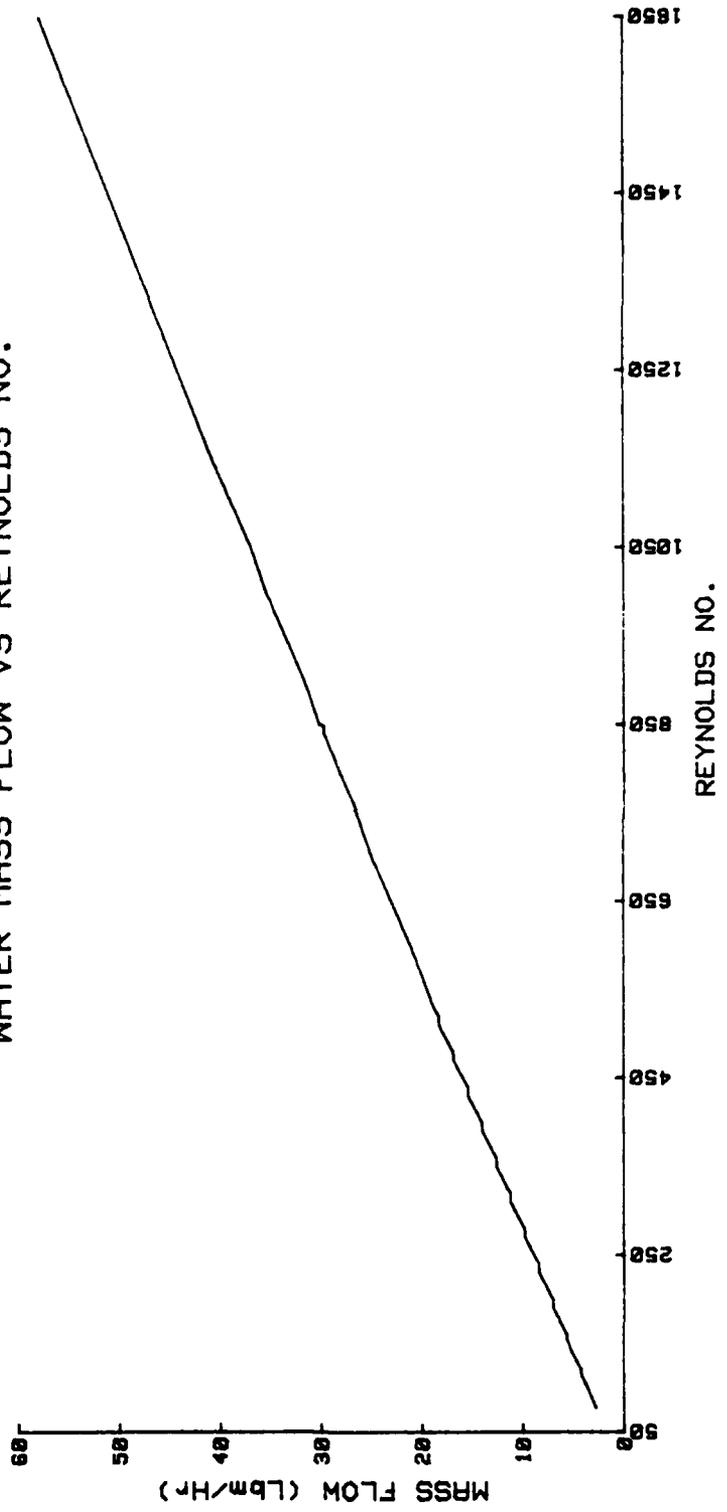
becomes simply $\quad Re_{out} = Re_{in} \dfrac{\mu_{INLET}}{\mu_{EXIT}} \quad .$

A logical expectation is the decrease in outlet superheat with an increase in mass flow rate. This trend is displayed in figure 42a. Point A corresponds to a Reynolds number of 310 and a superheat of 61°; whereas, point B, 380 and 91. Figures 43 and 44 show that there is very little difference in the water-side heat transfer coefficient for these two values of Reynolds numbers during the preheating of the water toward the saturation temperature. However, the value of h representative during superheating for Re = 380 is approximately twice the value of h for Re 310 (Figures 45 and 46).

The logical question to be raised at this point is why this difference exists. Outwardly, there definitely appears no logical explanation. Although not a specific part of the data set, the Reynolds number variation with axial position should have been plotted. It was observed on the CRT of the computer that in this "transition" region the Reynolds number gradient steadily increased along the tube for the inlet values of 310 and 380. This observation revealed the greatest magnitude of the Reynolds number was obtained at the point of 100% steam quality inside the tube.

For a Reynolds number of 310 the maximum value obtained was approximately 2100, whereas a Reynolds number of 380 the maximum value obtained was approximately 3300. The effect on heat transfer is observed as the water-side tube flow transverses from the laminar to turbulent regime, with the steam

95

1.0

1.1

1.25   1.4   1.6

2.8   2.5

3.2   2.2

3 6

4 0   2.0

1.8
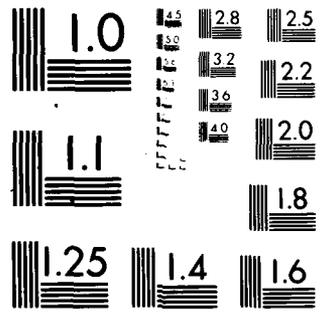
MICROCOPY RESOLUTION TEST CHART
NATIONAL BUREAU OF STANDARDS 1963 A

SUPERHEAT VS REYNOLDS NO.

FIGURE 42a

96
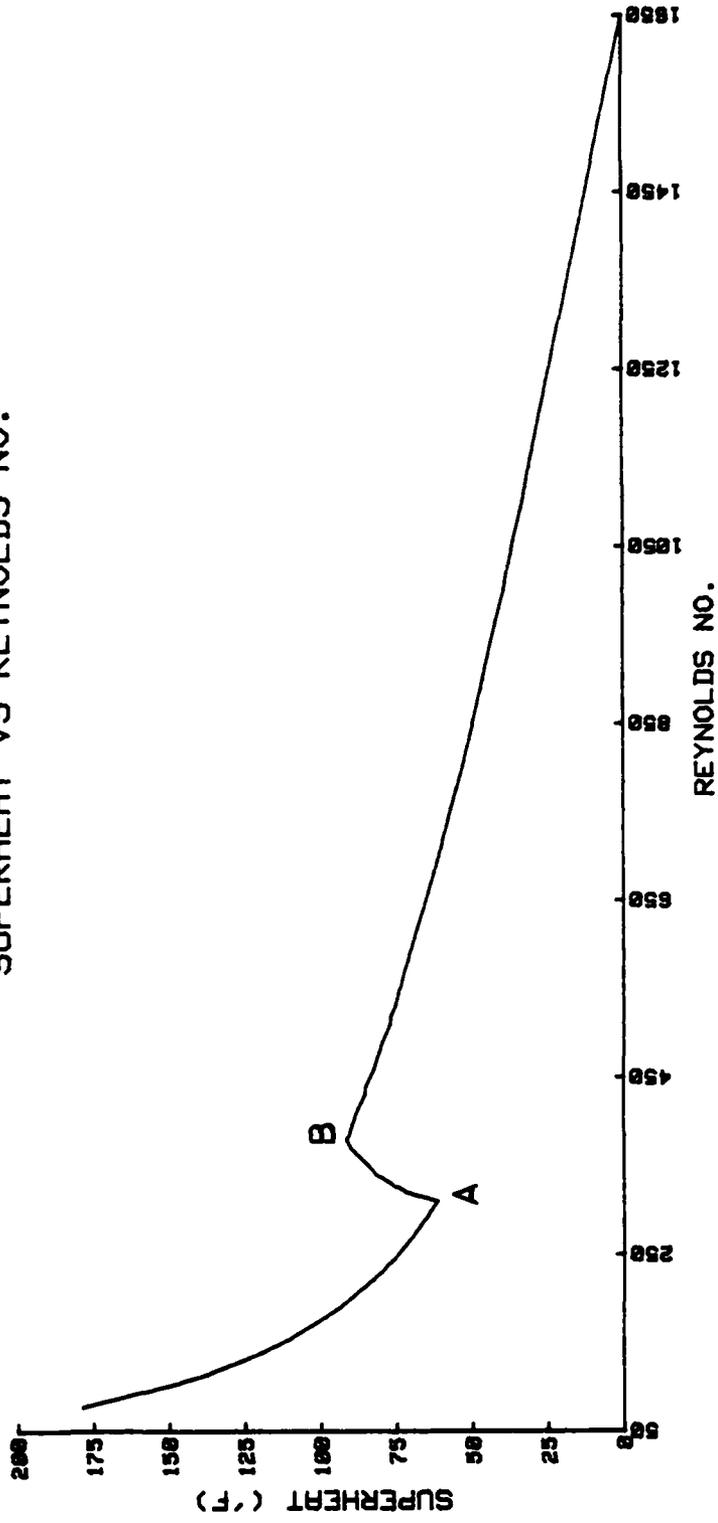
STEAM MASS FLOW VS SUPERHEAT
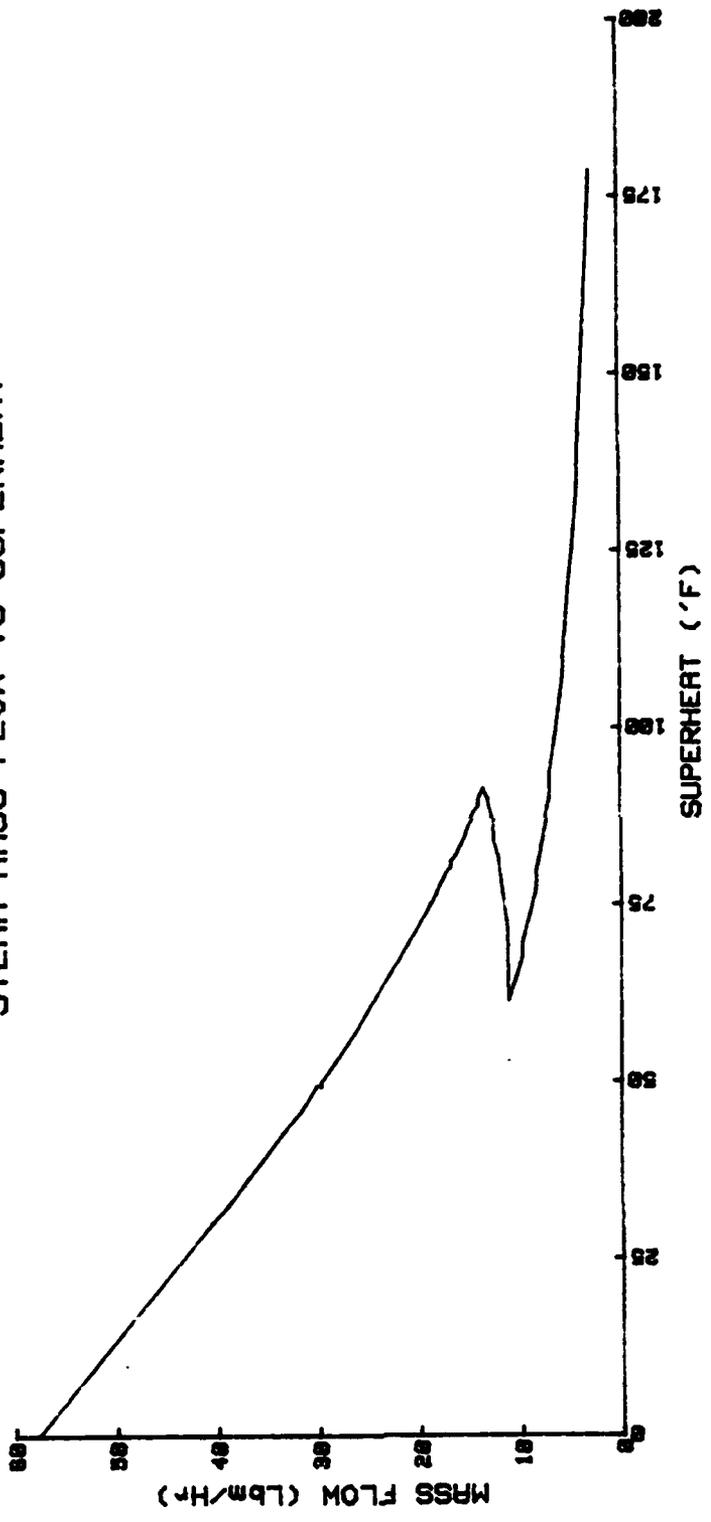
FIGURE 42b

REYNOLDS NO. = 310

FIGURE 43

98
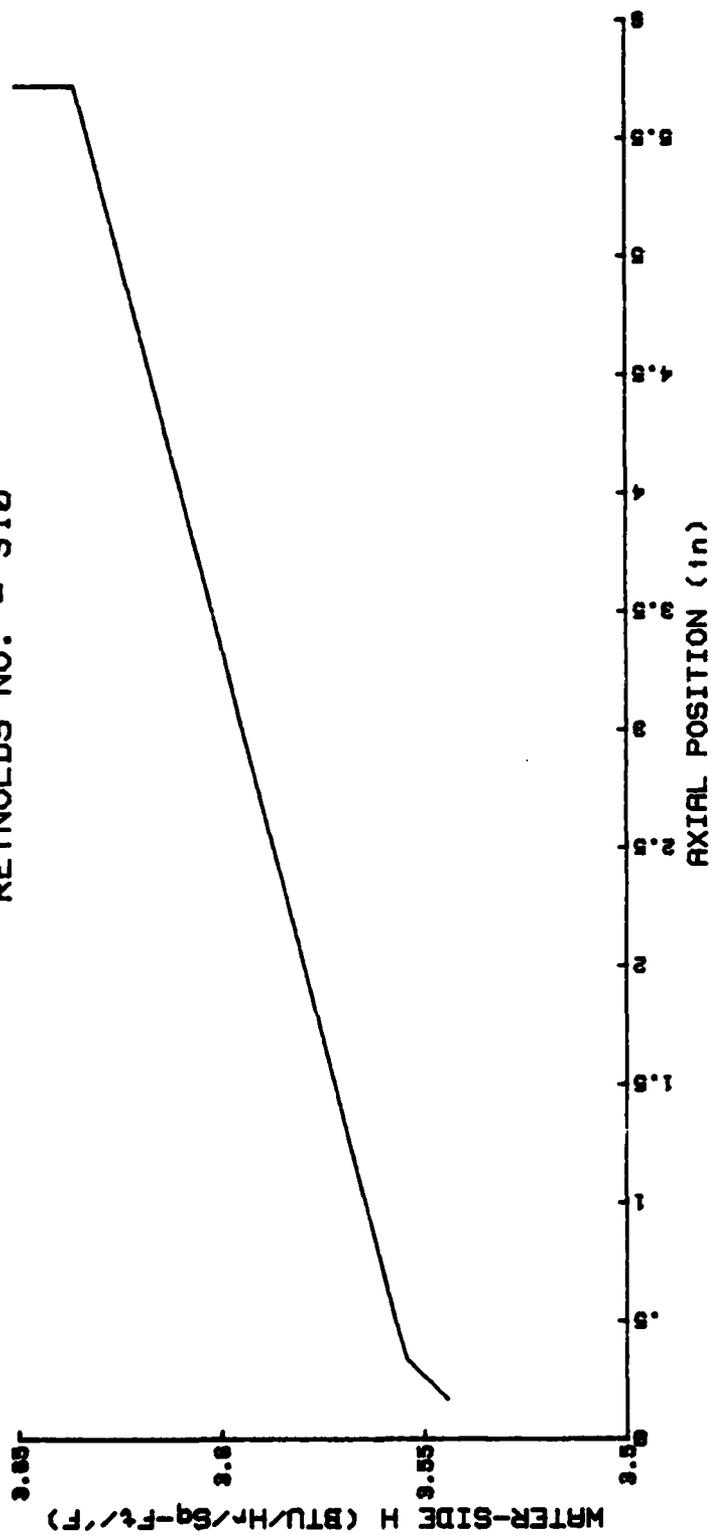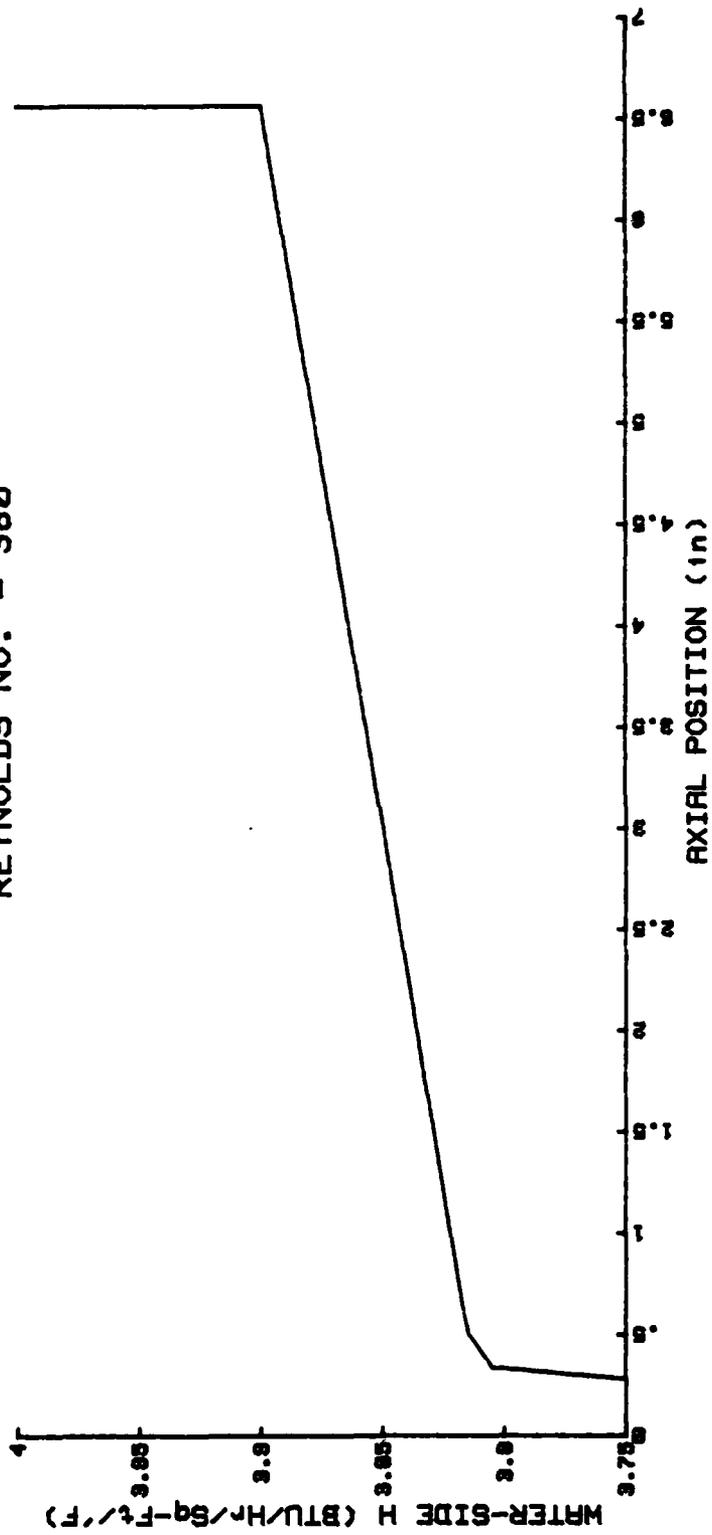
REYNOLDS NO. = 380

FIGURE 44

REYNOLDS NO. = 310

WATER-SIDE H (BTU/Hr/Sq-Ft/°F)

AXIAL POSITION (in)
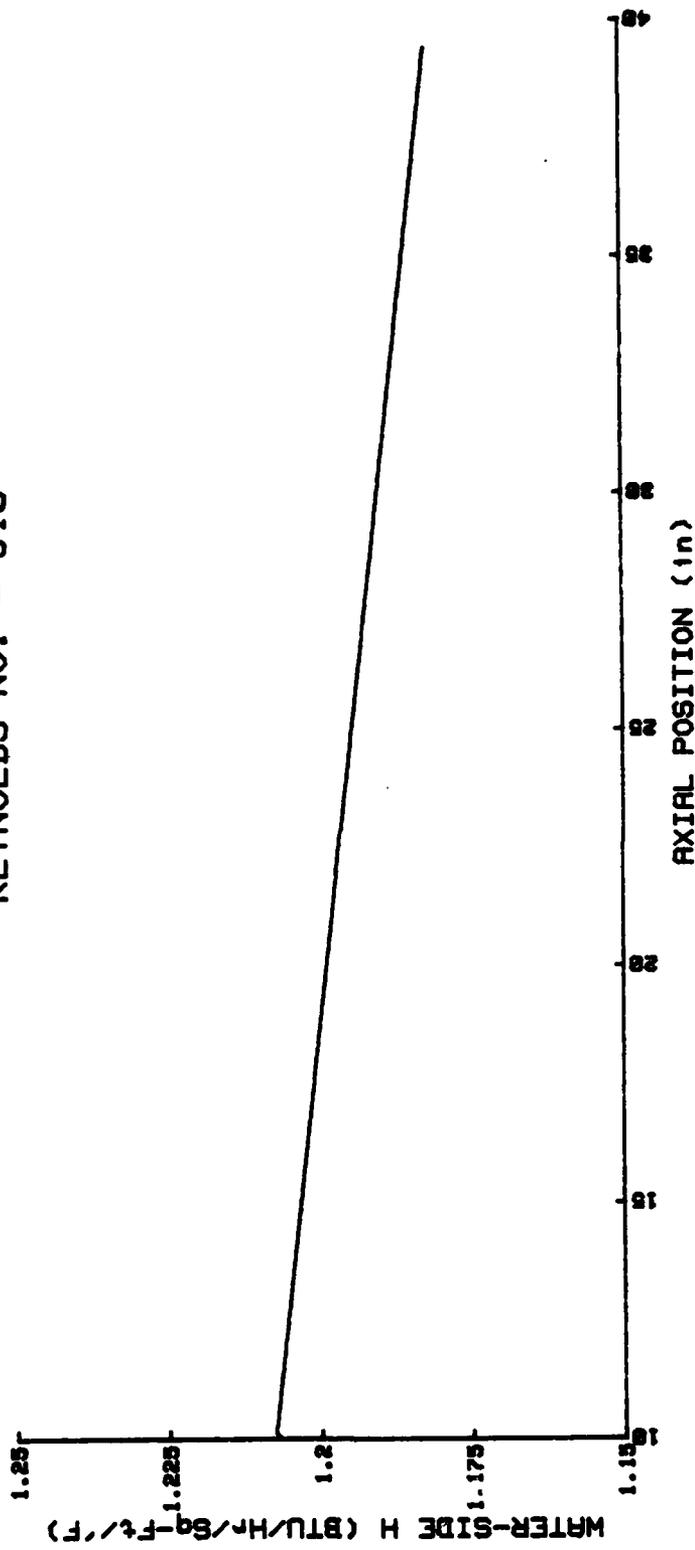
FIGURE 45

REYNOLDS NO. = 380
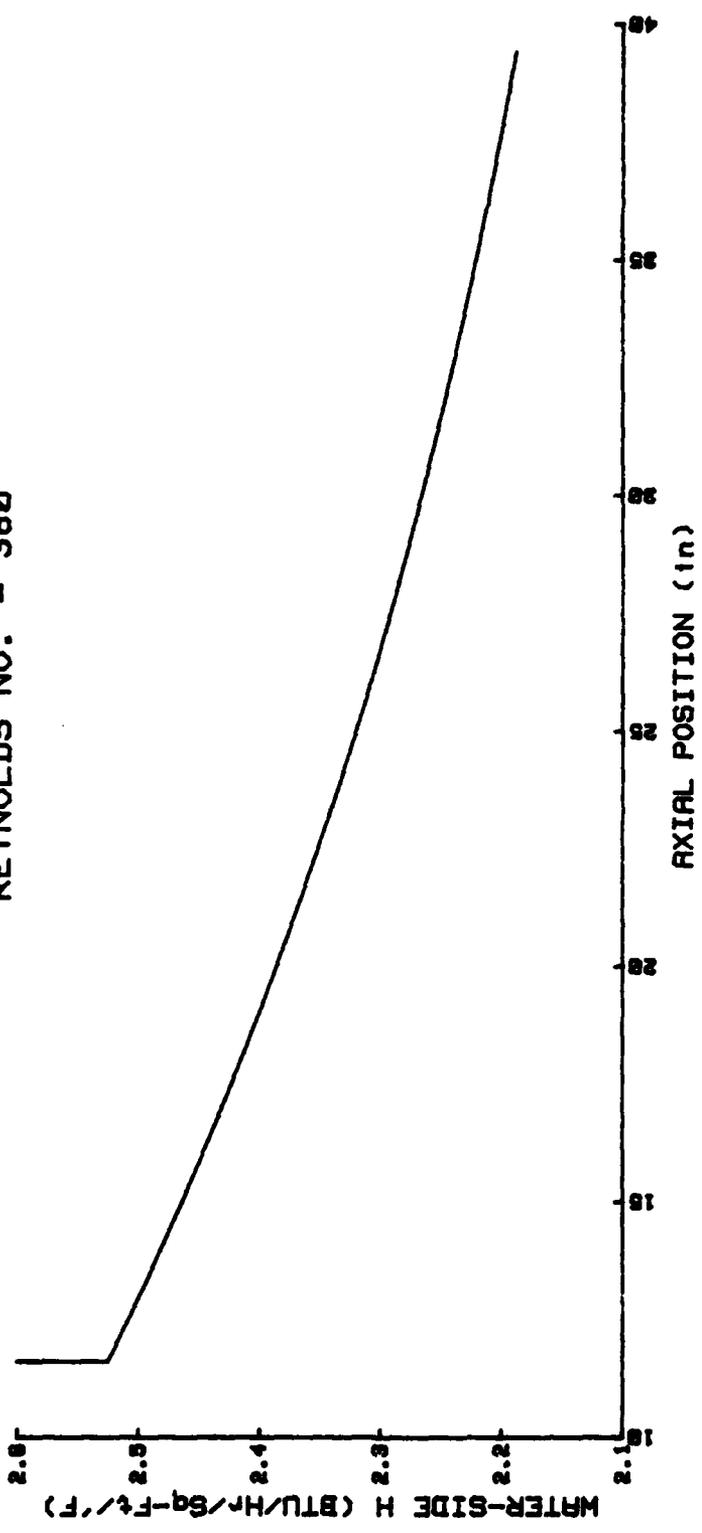
FIGURE 46

101

REYNOLDS NO. = 310

WATER BULK TEMPERATURE ('F)

AXIAL POSITION (in)

FIGURE 47

102

REYNOLDS NO. = 380

FIGURE 48

103

FIGURE 49

104

LOCATION OF 100 % QUALITY

REYNOLDS NO.

AXIAL POSITION (in)

FIGURE 50

105
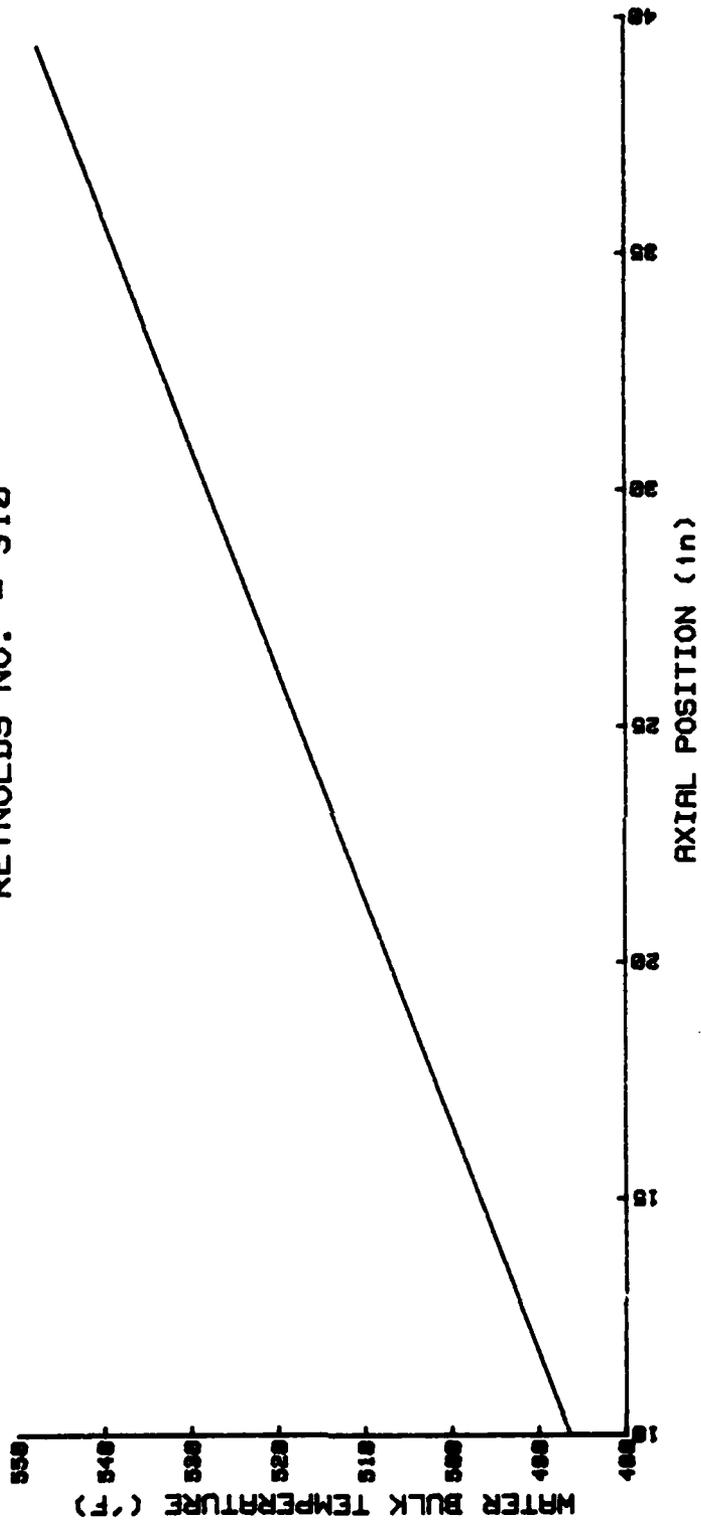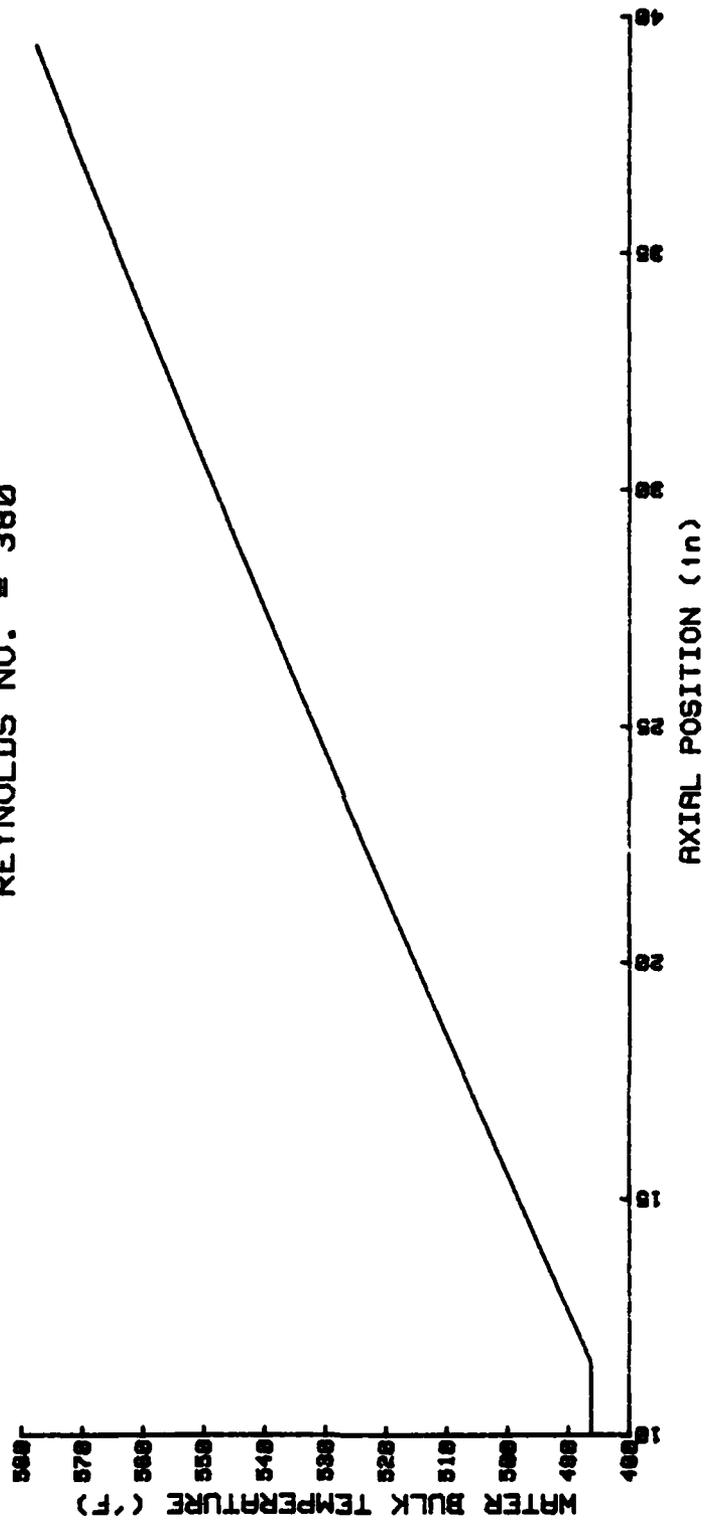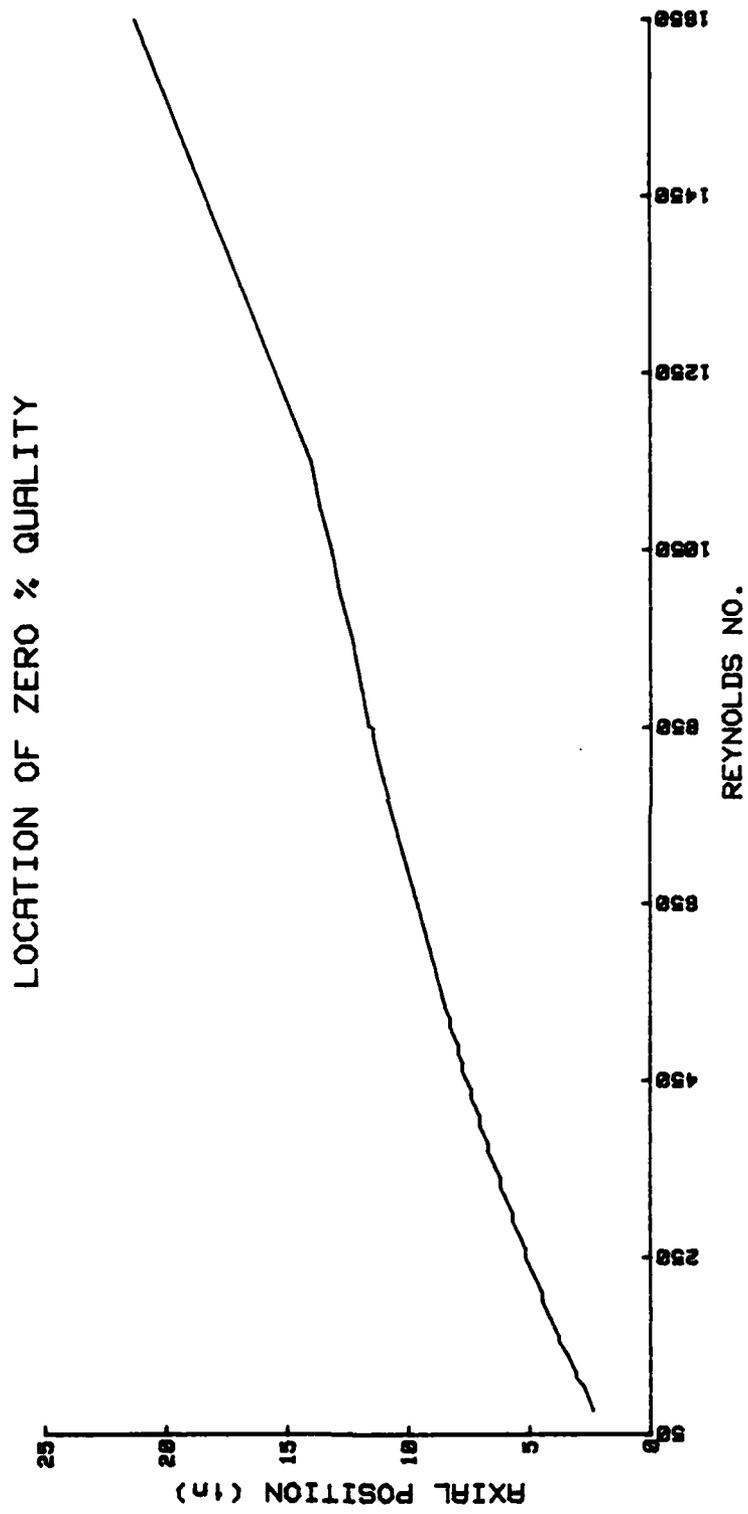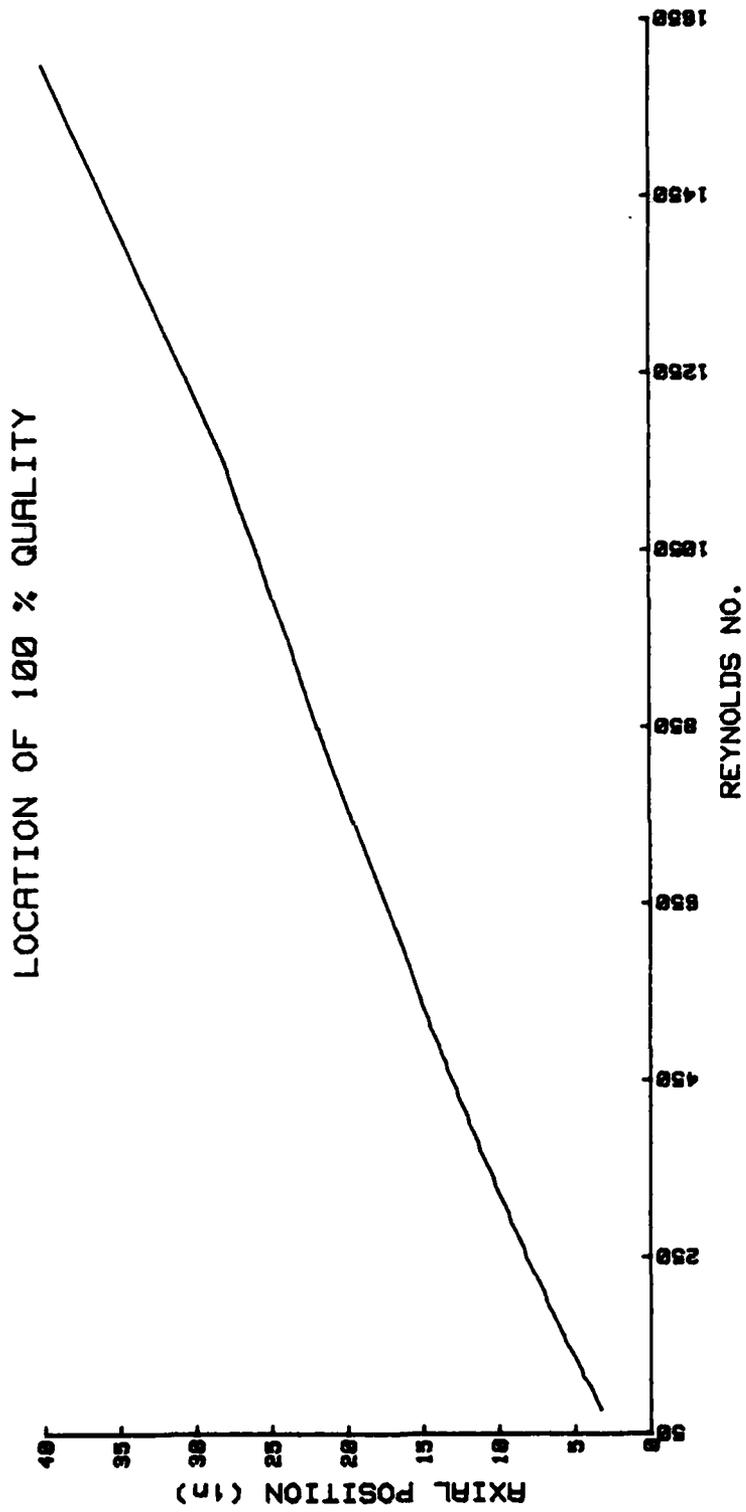
bulk temperature variation reflecting this fact. In figure

47, the change in steam temperature is approximately 24.4°F/FT

whereas in Figure 48, it is 38.6°F/FT.

Figures 49 and 50 show the axial location of initiation of

boiling (zero percent quality) and that of complete evapora-

tion (100 percent quality) as a function of the inlet Reynolds

number.

The target of 50 degrees superheat was obtained with an

inlet Reynolds number of 840.

Table I. HEAT BALANCE RESULTS

Inlet water-side Reynolds number	840	
Outlet water-side Reynolds number	5180.19	
Overall water mass flow rate	29.648202	lbm/hr
Total overall heat addition rate	24621.249	BTU/hr
The inlet water temperature	436.21	$^\circ$F
The saturation temperature	486.21	$^\circ$F
The outlet steam temperature	536.22	$^\circ$F
The amount of superheat	50.01	$^\circ$F
The inlet enthalpy	414.867	BTU/lbm
The outlet enthalpy	1245.314	BTU/lbm

Figures 51 through 59 graphically represent the data

obtained from the model simulation for Re = 840.

These figures show the axial distribution of the associated variables under the prescribed design conditions for the segmented fin profile.

Table 2: TUBE/FIN DIMENSIONS AND AREAS

The selected fin profile is the segmented-fin

The outside tube diameter	2	Inches
The inside tube diameter	1.86	Inches
The tube length	39.37	Inches
The total outside tube area	1.71889	Square Feet
The total inside tube area	1.59857	Square Feet
The fin root diameter	2	Inches
The number of 'fins' per inch	5.94	
The number of 'segments' per fin	38	
The fin height	1.015	Inches
The fin thickness	.048	Inches
The transverse tube pitch	4.5	Inches
The longitudinal tube pitch	2.25	Inches
The fin outside diameter	4.03	Inches
The fin segment width	.17	Inches
The length of cut from fin tip	.82	Inches
The total integer number of fins	234	Fins
Actual tube length will be	39.394	Inches
The fin thermal conductivity	144	BTU/Hr.Ft./F
The tube thermal conductivity	20	BTU/Hr.Ft./F

Calculated heat transfer areas in square feet:

Total heat exchanger frontal area	39.394
Total fin area	26.950951
Total bare tube area	1.228802
Total outside tube area	28.179753
Single fin area	.115175
Outside 'elemental' area	.120426
Inside 'elemental' area	.006831
'Elemental' frontal area	.005261
'Elemental' blocked area	.003015
'Elemental' minimum gas-flow area	.002246

This data shows what would reasonably be expected. That is, relatively low values for heat transfer coefficients and heat flux for the preheat and superheating sections and high values for the boiling section.

D. CONCLUSION

The short straight tube model and the accompanying graphical data provide the basic framework for additional study of the stagnation, or S.S.T boiler. This model, as formulated, provides a fairly reasonable estimate of the possible performance for the fin-tube configuration considered. Performance estimates are encouraging even considering the assumptions involved.

REYNOLDS NO. = 840

GAS BULK TEMPERATURE (°F)

AXIAL POSITION (in)
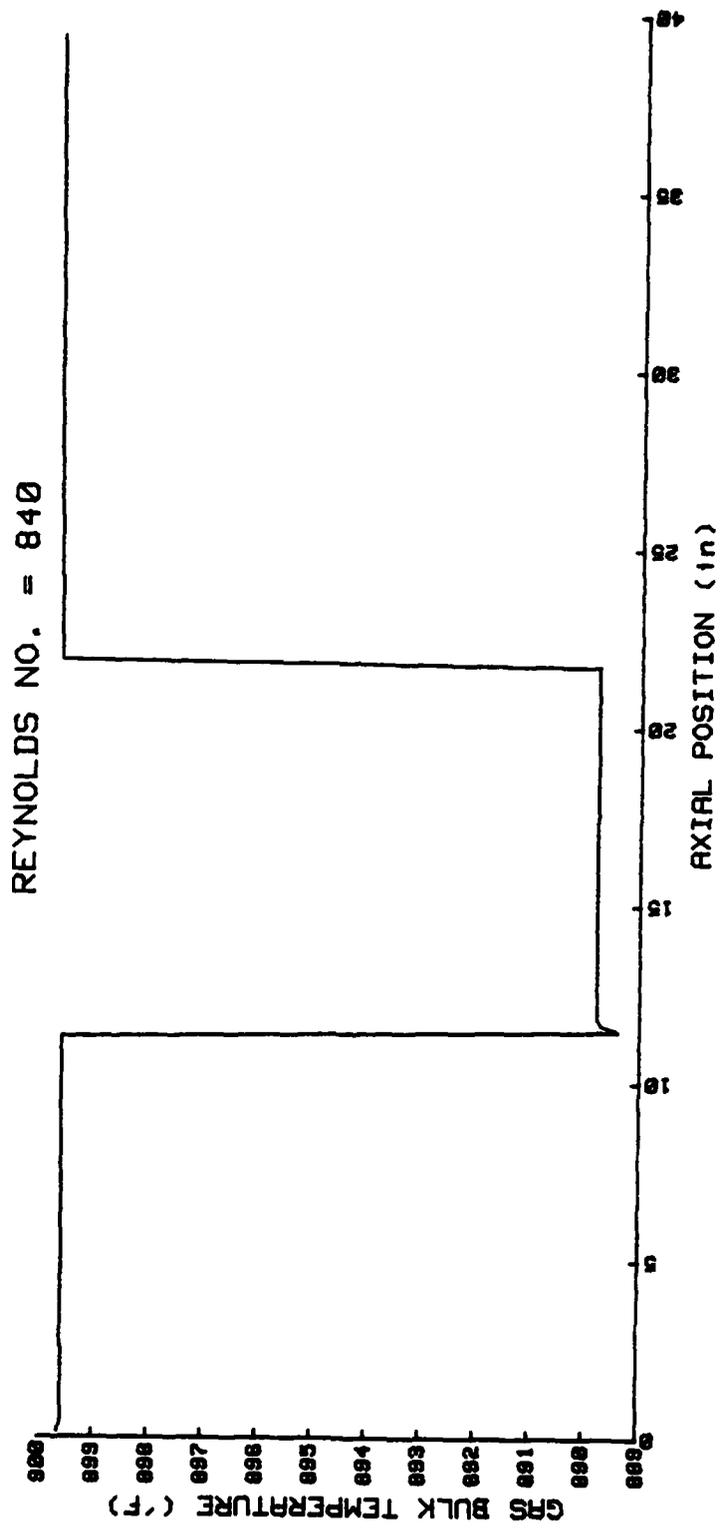
FIGURE 51a

REYNOLDS NO. = 840

FIGURE 51b

110

REYNOLDS NO. = 840

GAS BULK TEMPERATURE (°F)

AXIAL POSITION (in)
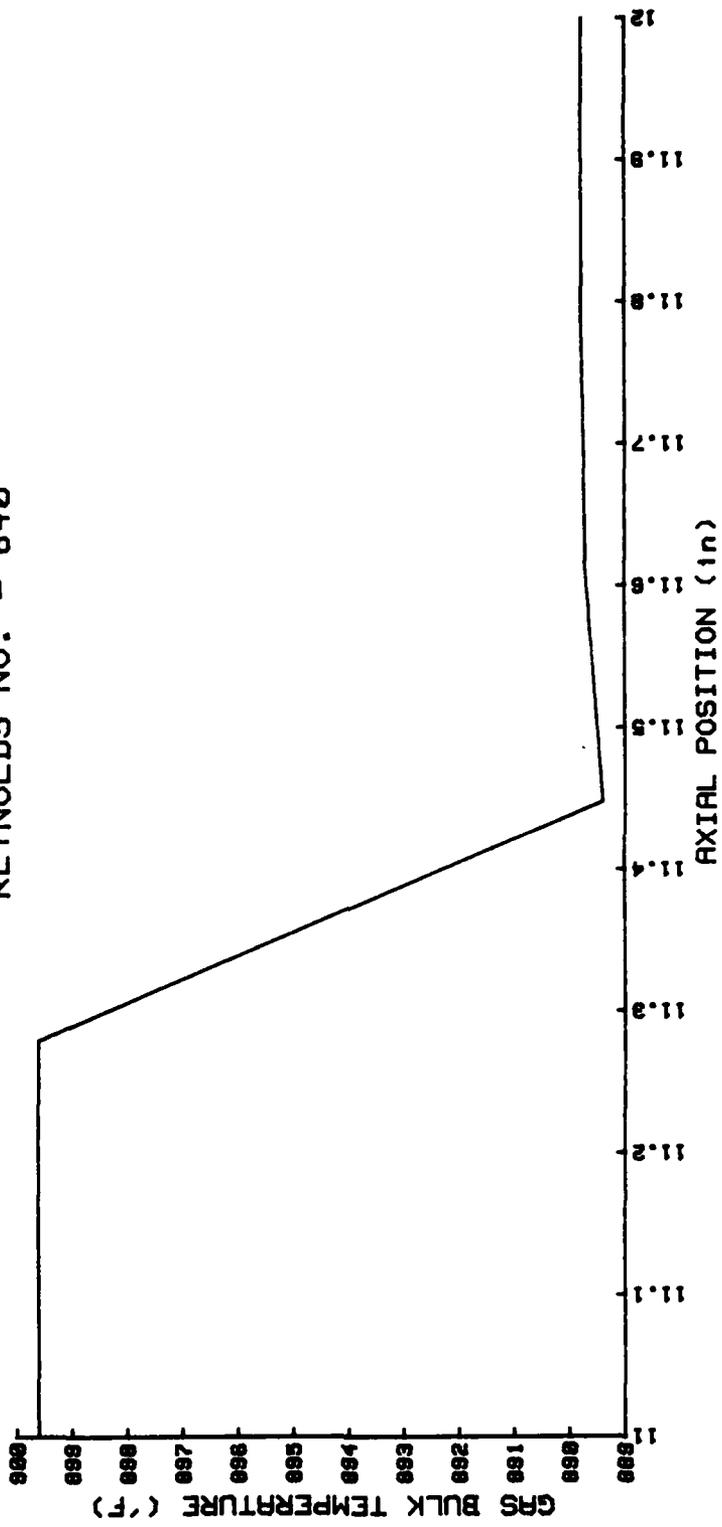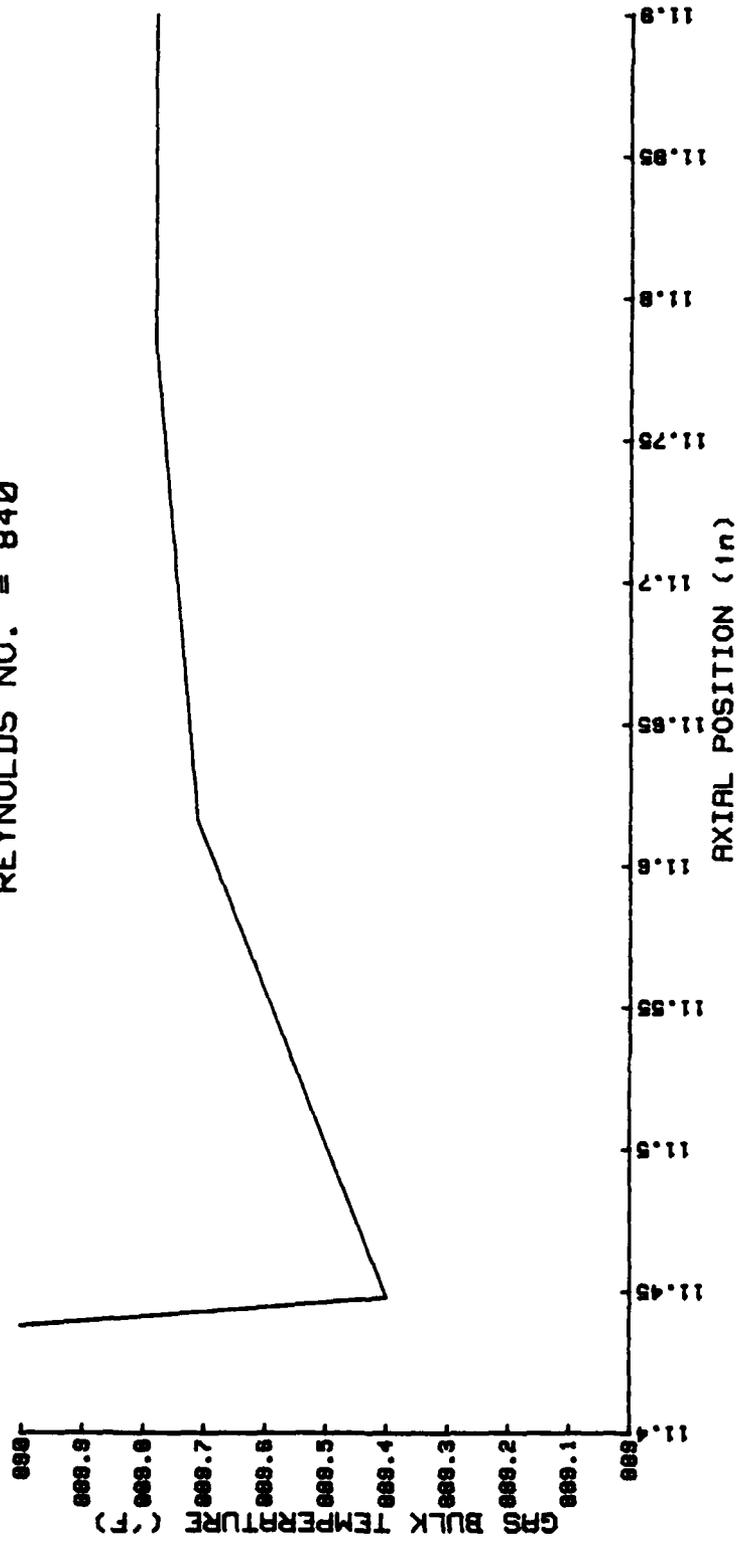
FIGURE 51c

REYNOLDS NO. = 840
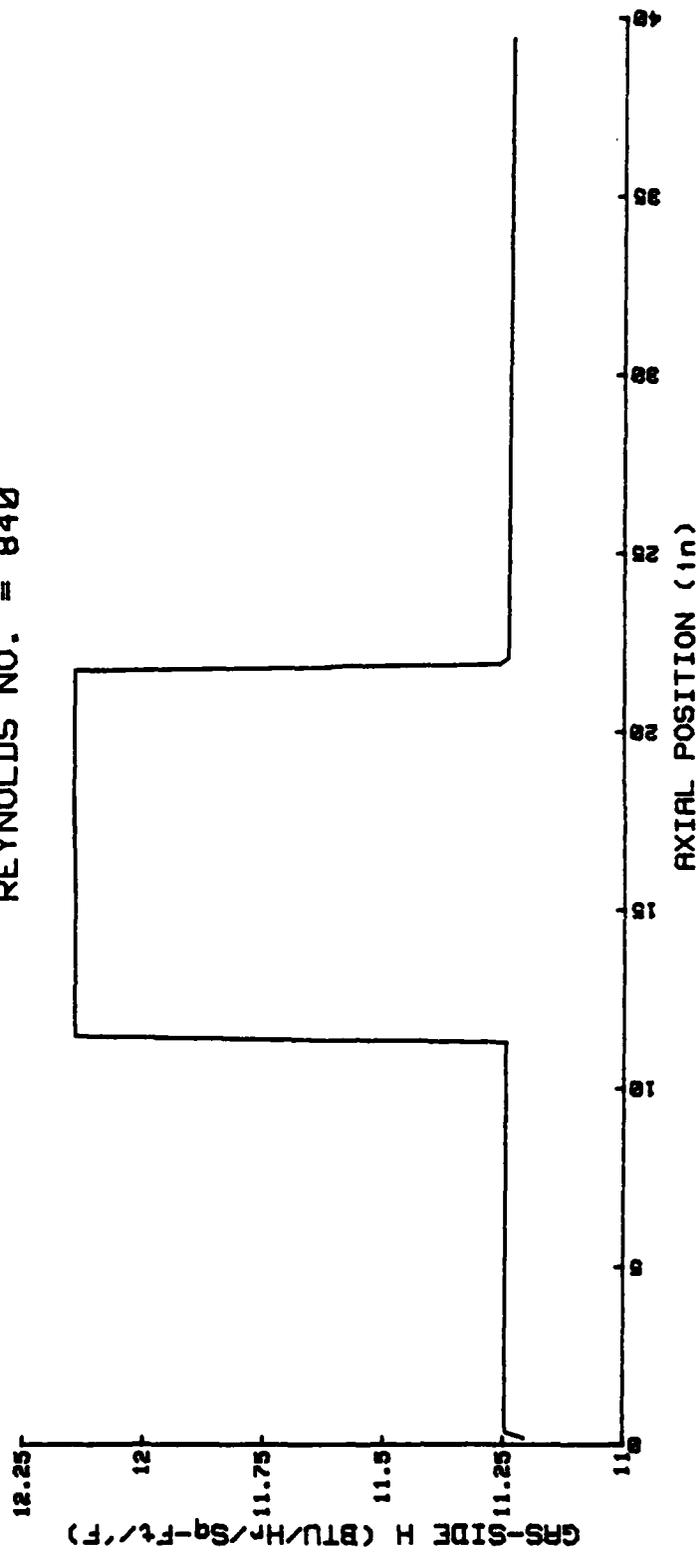
FIGURE 52a
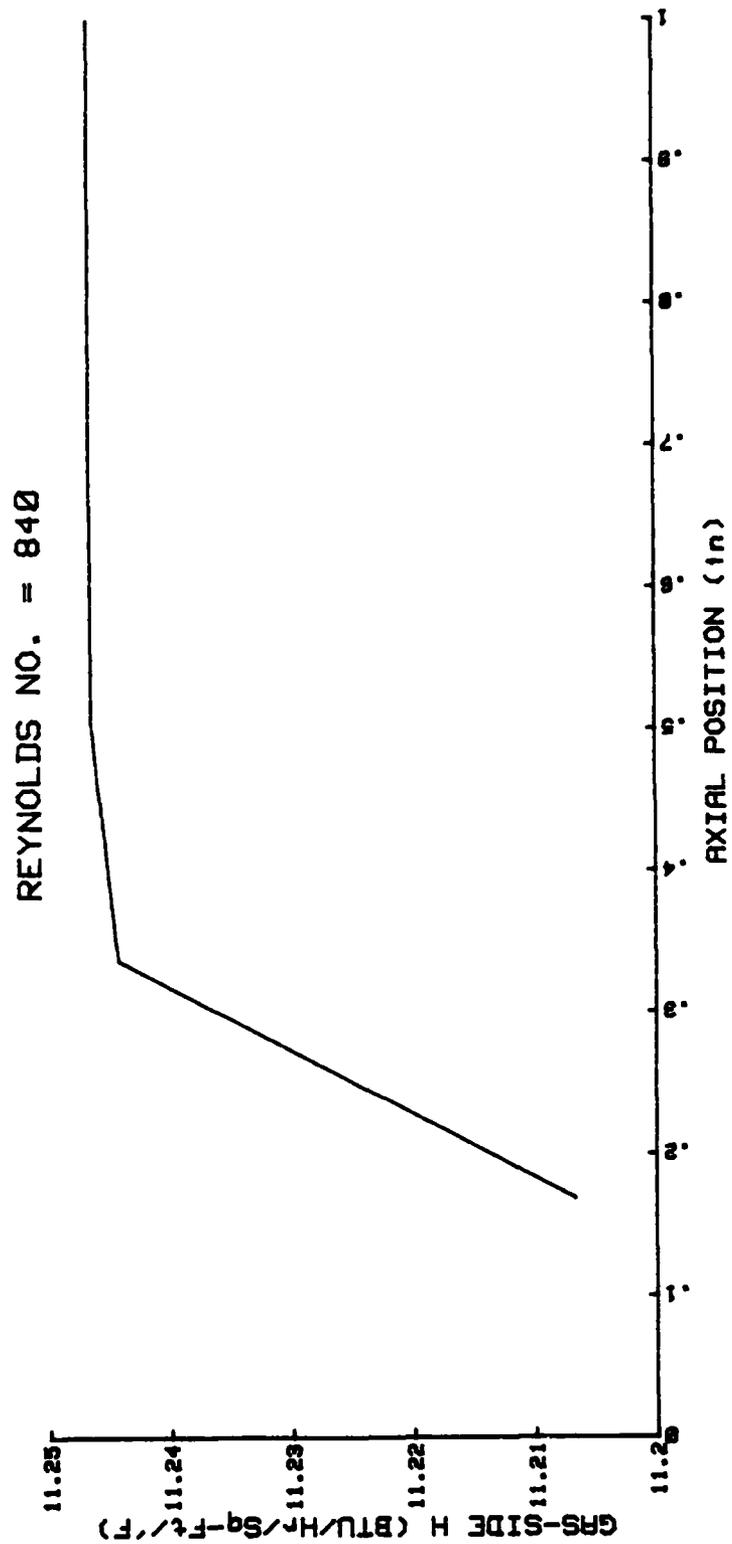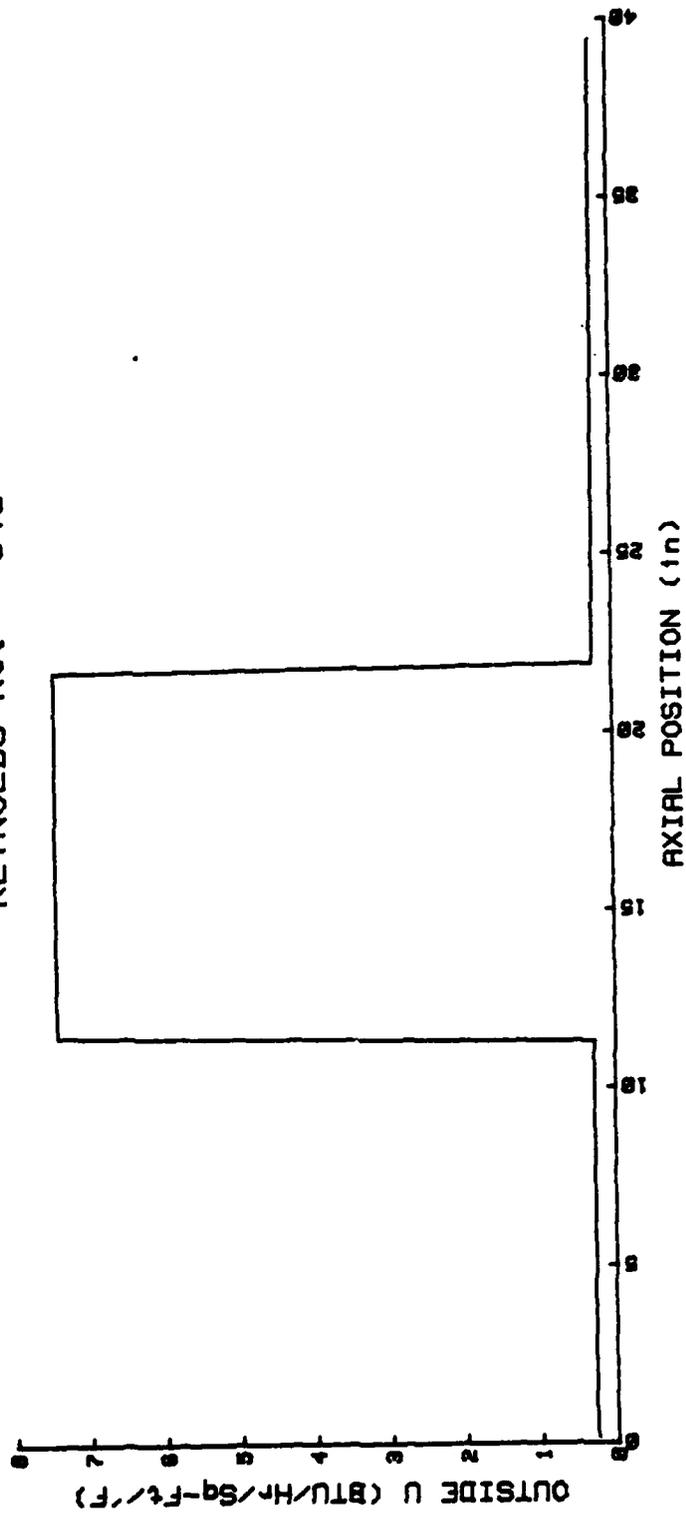
FIGURE 52b

113

REYNOLDS NO. = 840

GAS-SIDE H (BTU/Hr/Sq-ft/°F)

AXIAL POSITION (in)

FIGURE 52c

114

REYNOLDS NO. = 840

OUTSIDE U (BTU/Hr/Sq-Ft./°F)

AXIAL POSITION (in)

FIGURE 53a

115

REYNOLDS NO. = 840

FIGURE 53b

116

REYNOLDS NO. = 840

AXIAL POSITION (in)

OUTSIDE U (BTU/Hr/Sq-Ft/°F)

FIGURE 53c

117

REYNOLDS NO. = 840

FIGURE 54a

118
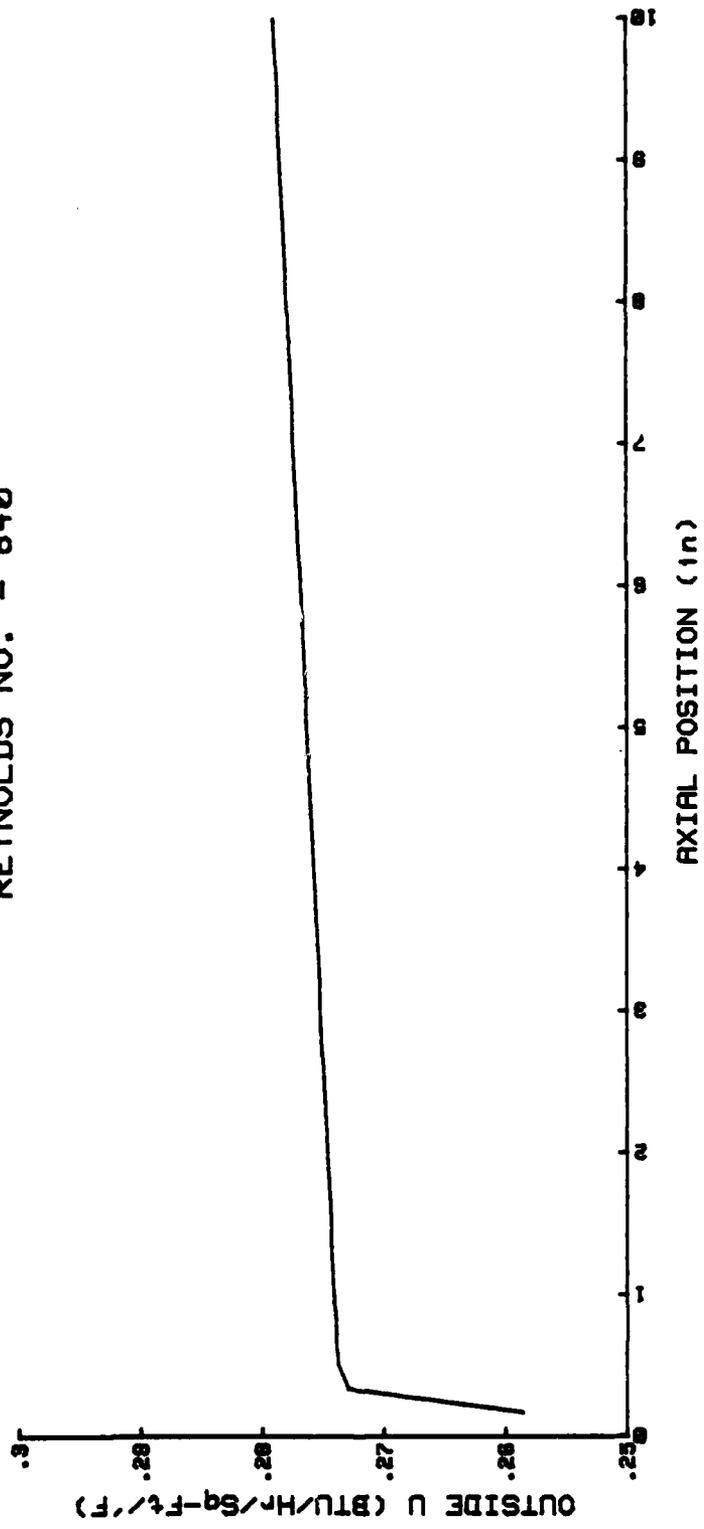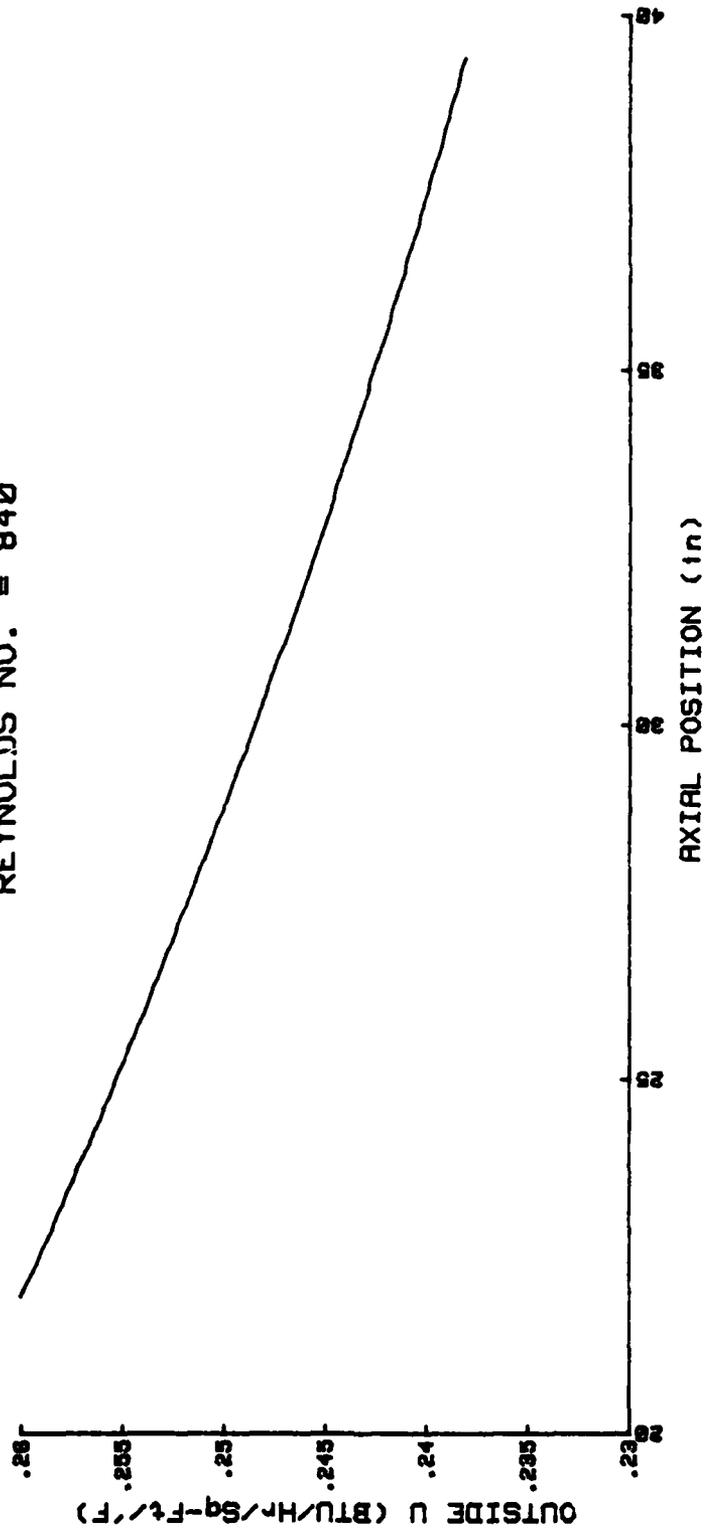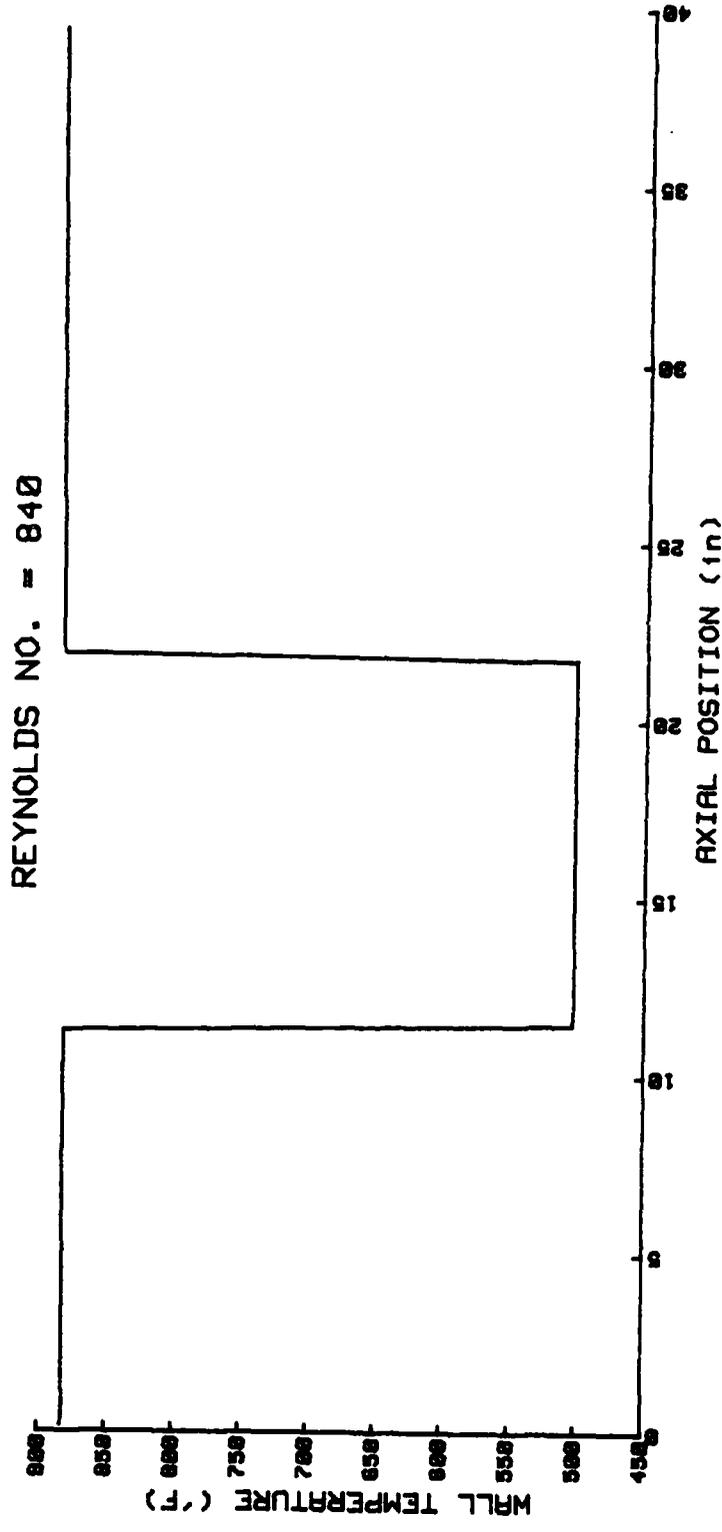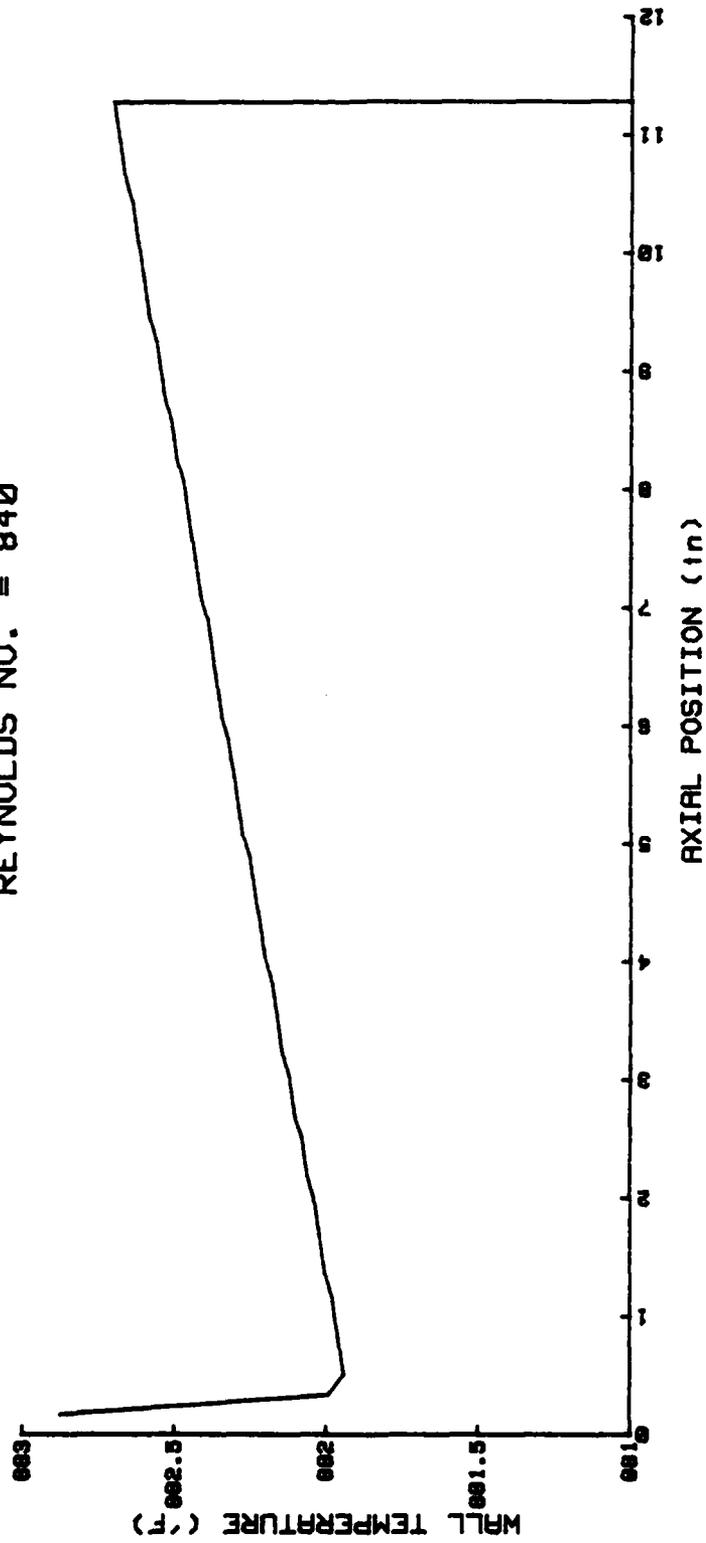
REYNOLDS NO. = 840

FIGURE 54b

REYNOLDS NO. = 840

WALL TEMPERATURE (°F)

AXIAL POSITION (in)

FIGURE 54c

120
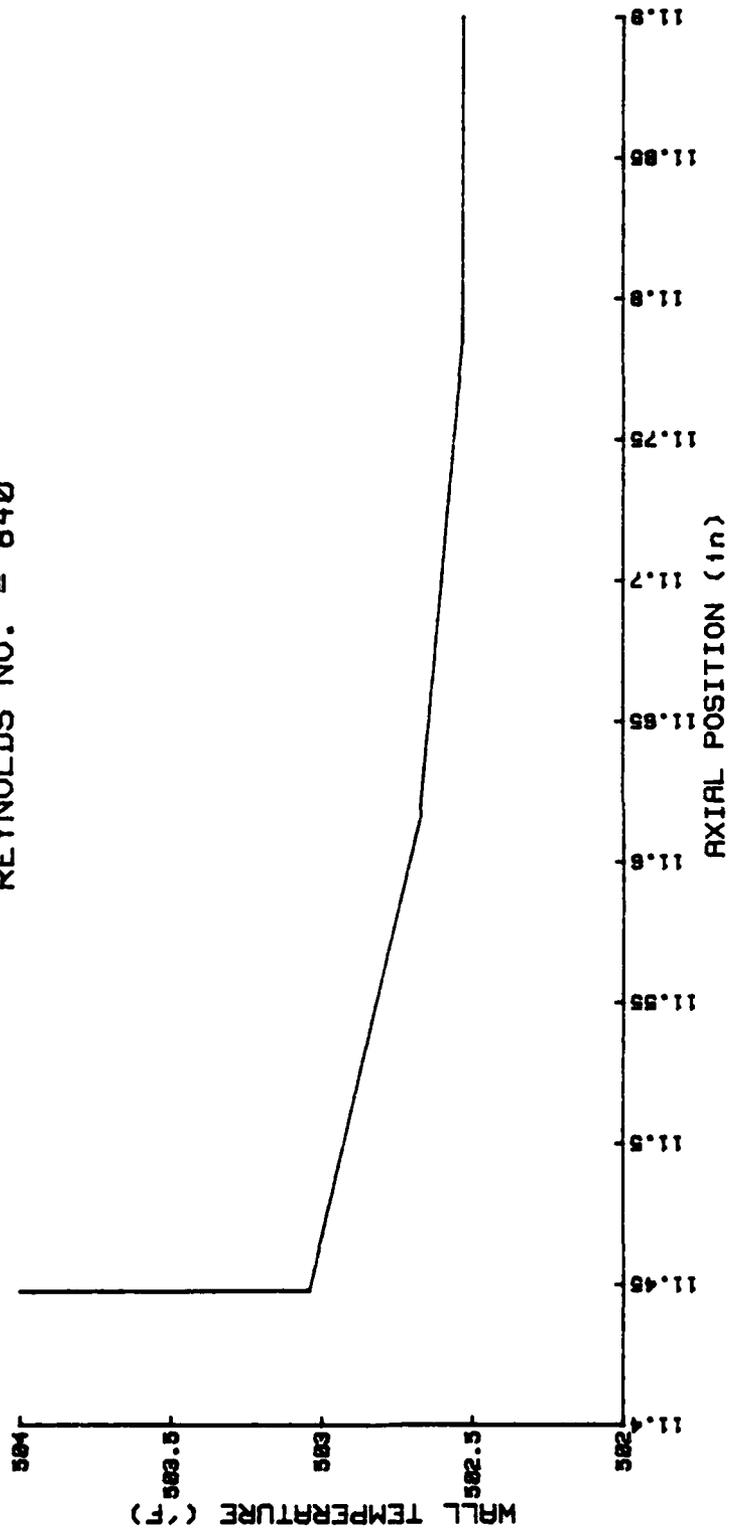
REYNOLDS NO. = 840

FIGURE 5.5a

121
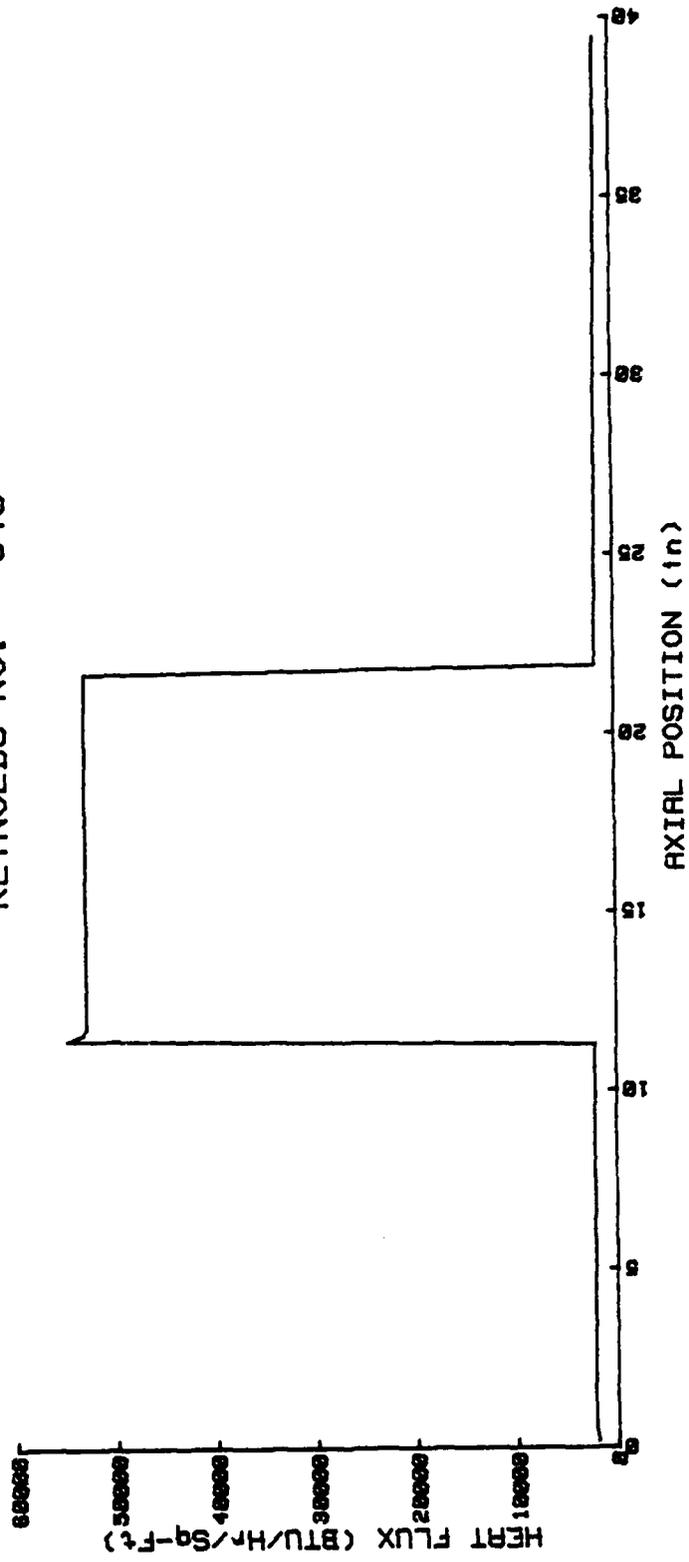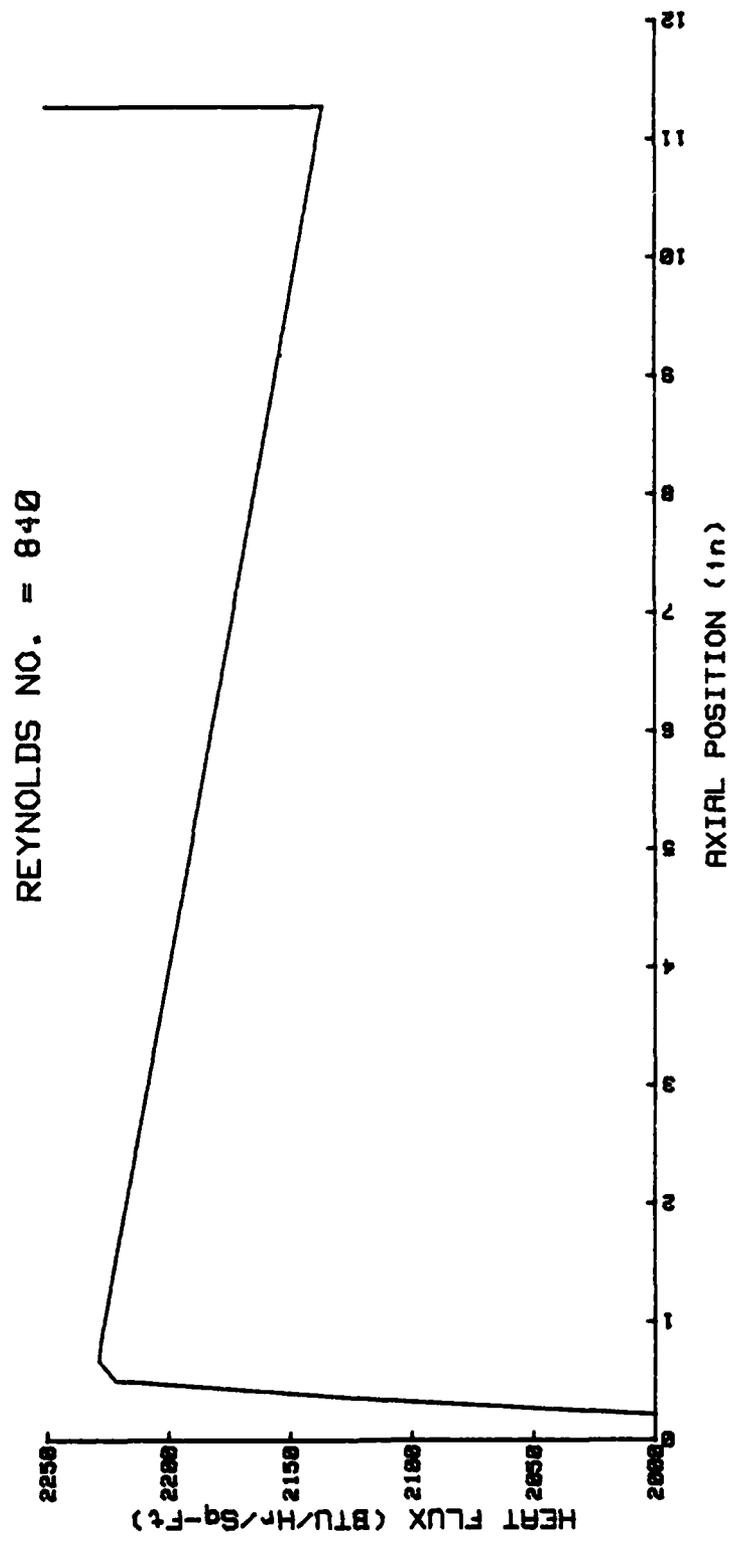
REYNOLDS NO. = 840

FIGURE 55b

REYNOLDS NO. = 840

FIGURE 55c

123

REYNOLDS NO. = 840

HEAT FLUX (BTU/Hr/Sq-Ft)

AXIAL POSITION (in)

FIGURE 55d

124
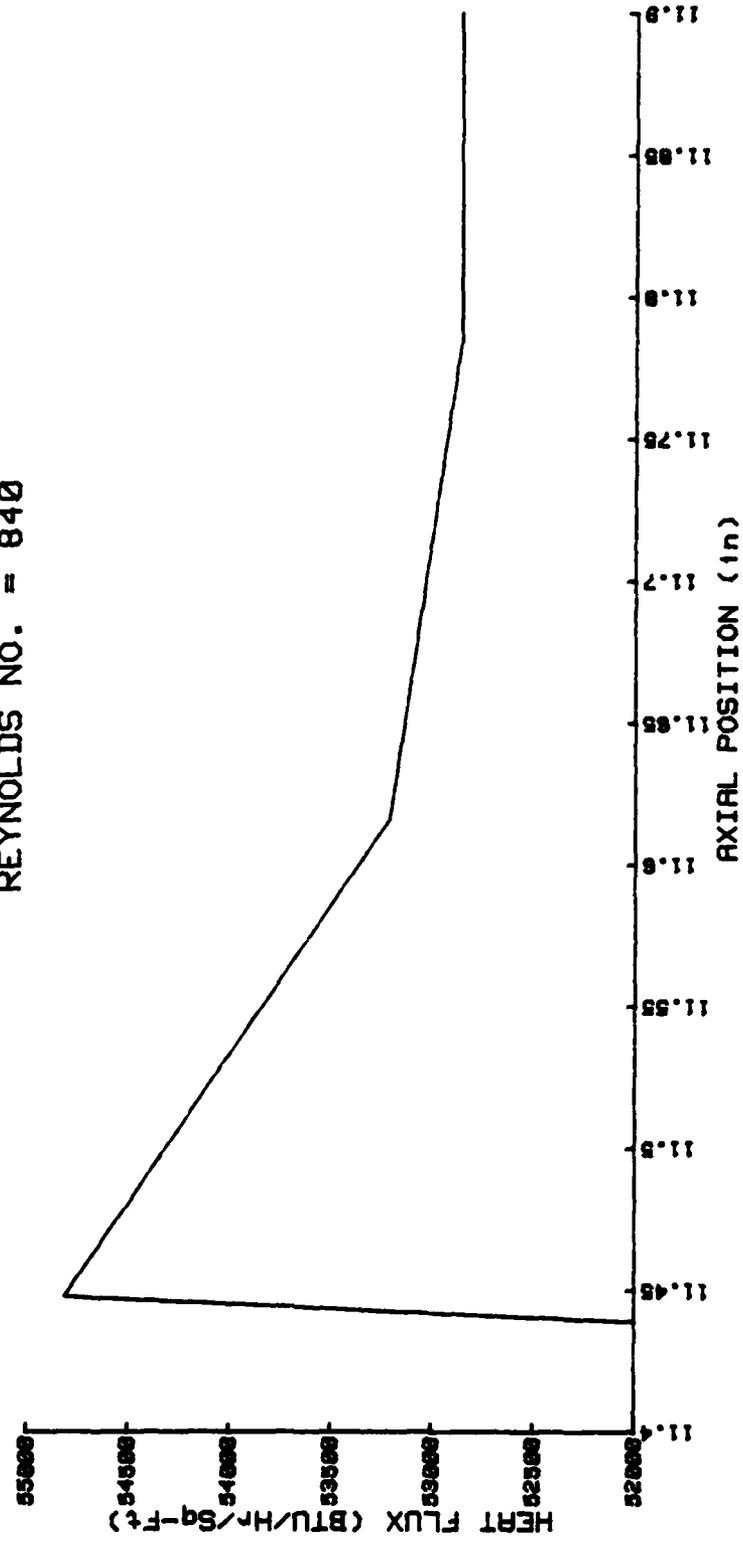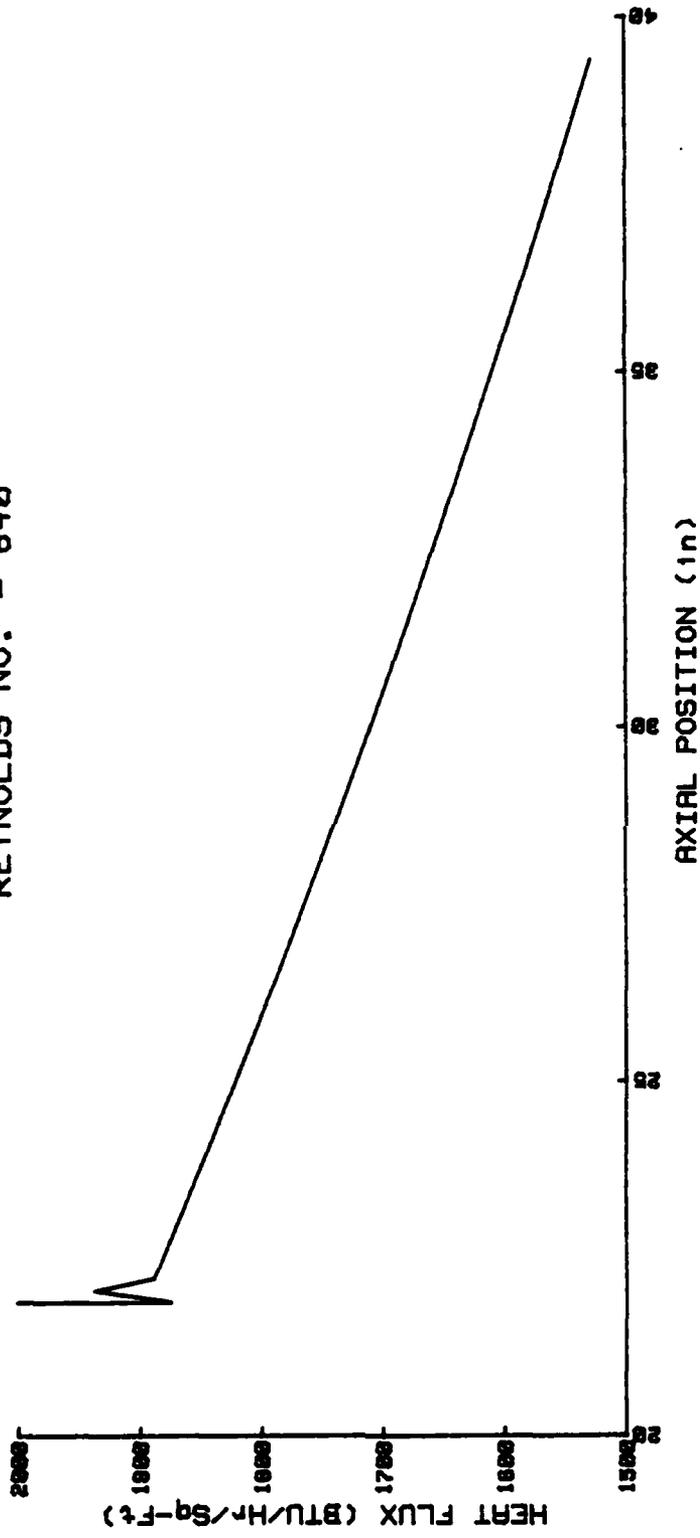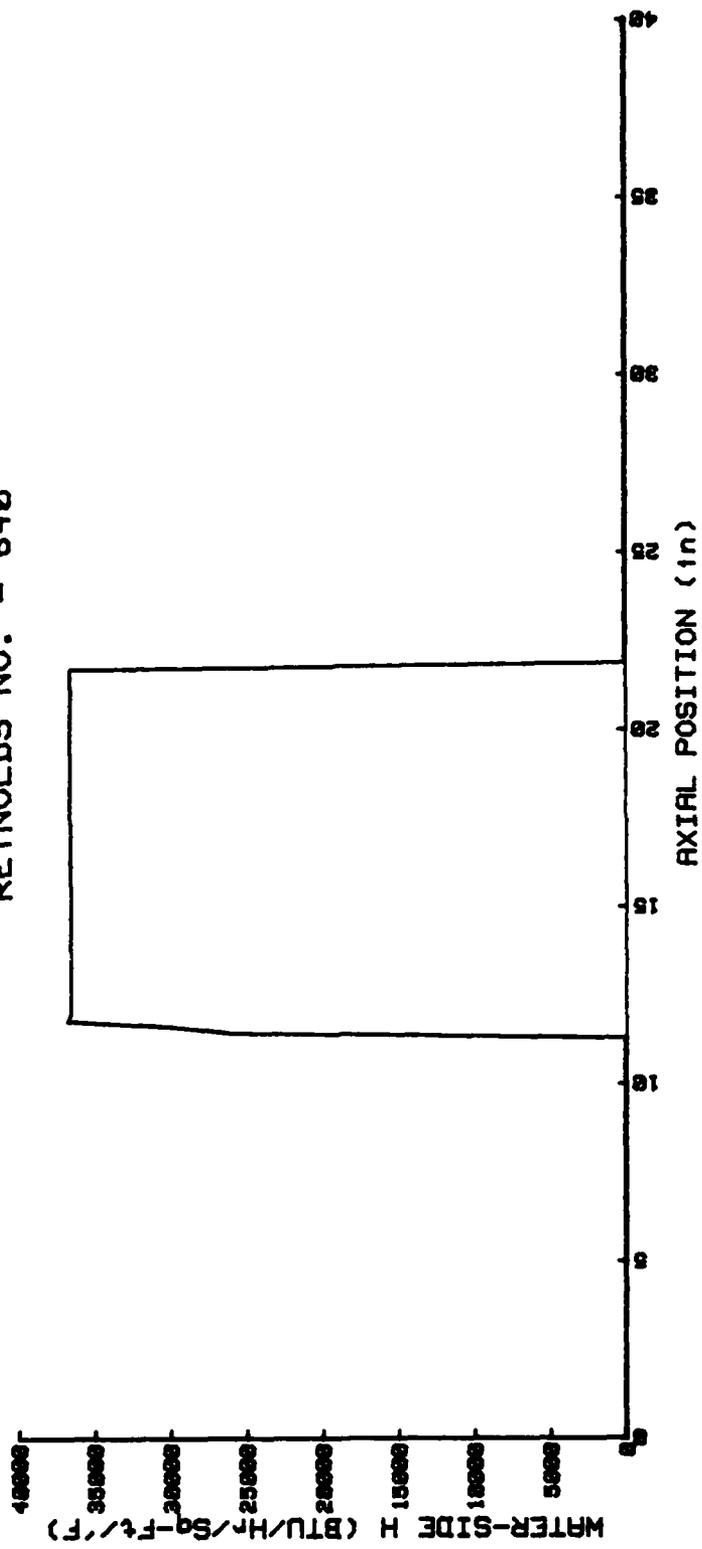
FIGURE 56a

REYNOLDS NO. = 840

FIGURE 56b

126
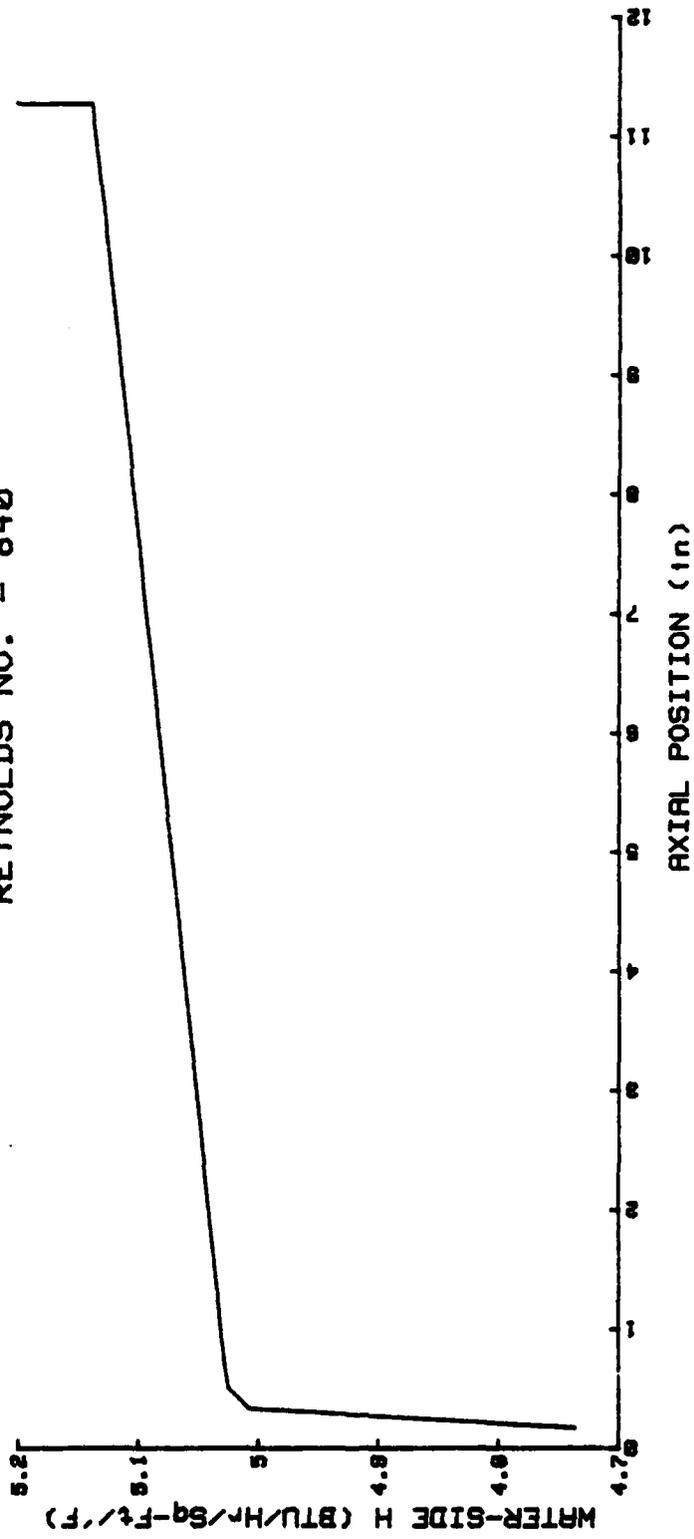
REYNOLDS NO. = 840

FIGURE 56c

127
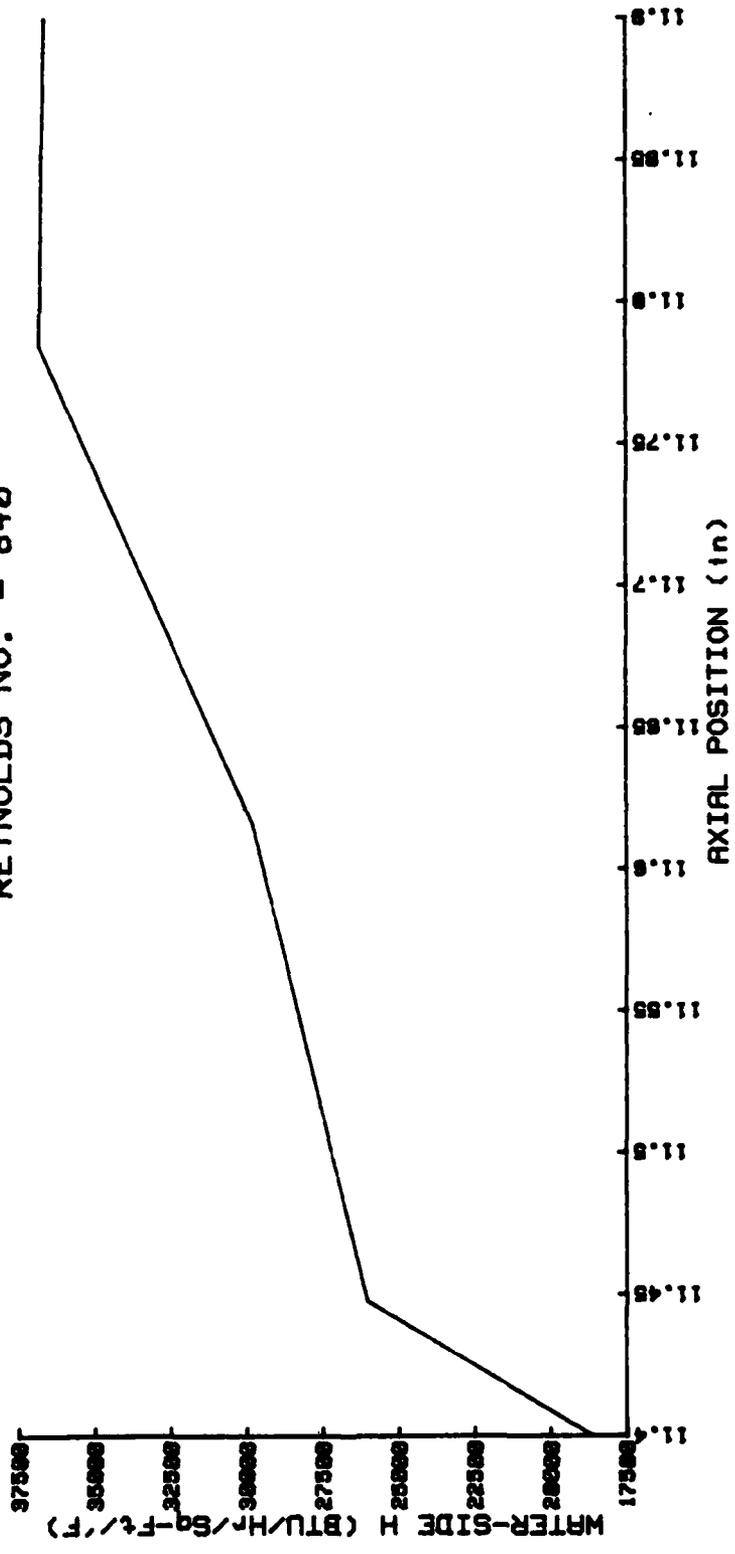
REYNOLDS NO. = 840

FIGURE 56d

128
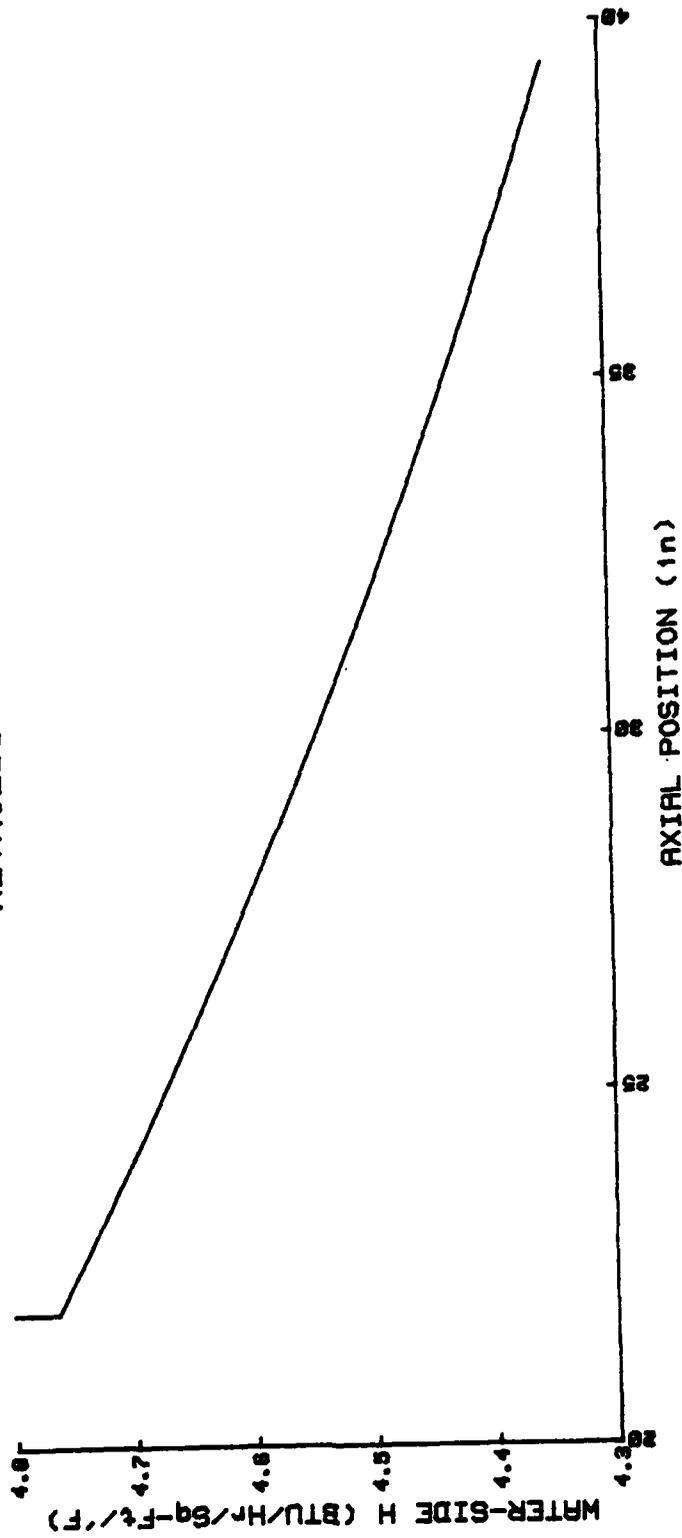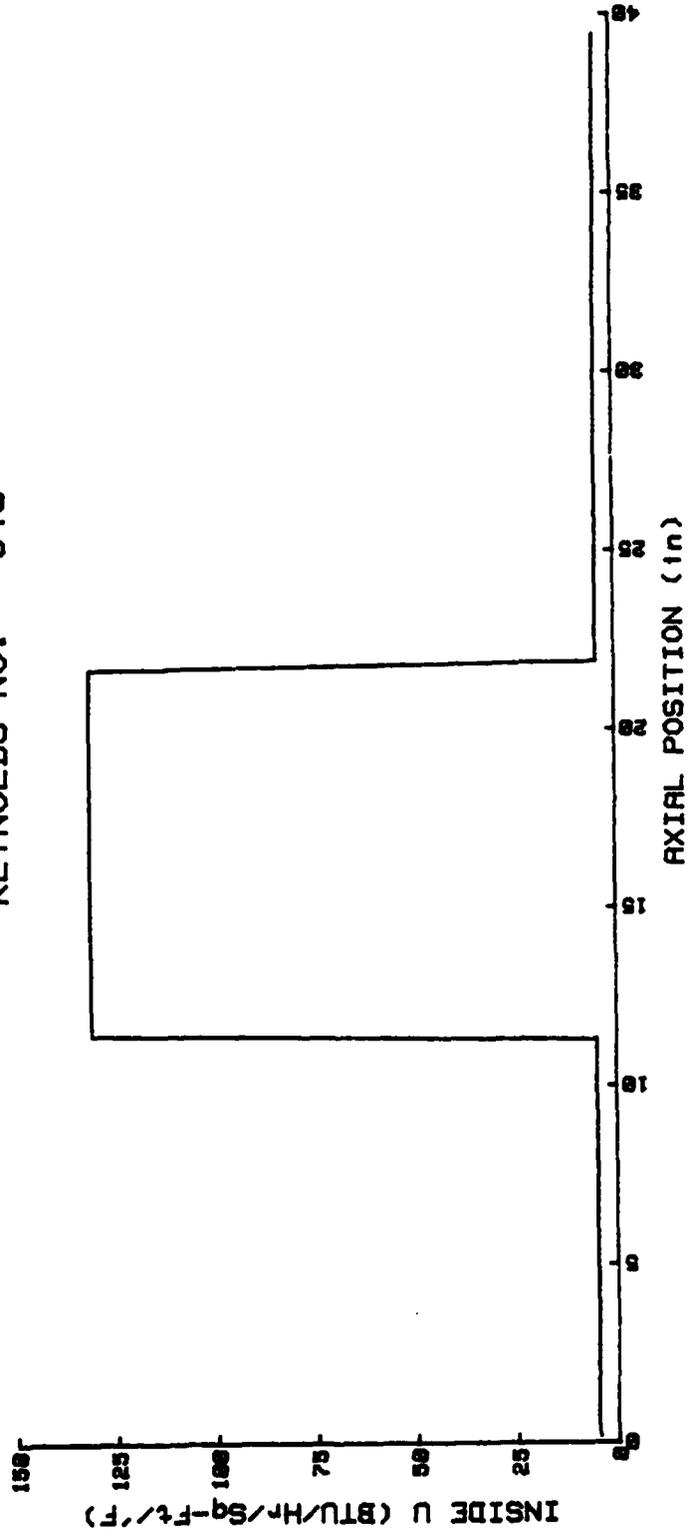
REYNOLDS NO. = 840

FIGURE 57a

REYNOLDS NO. = 840
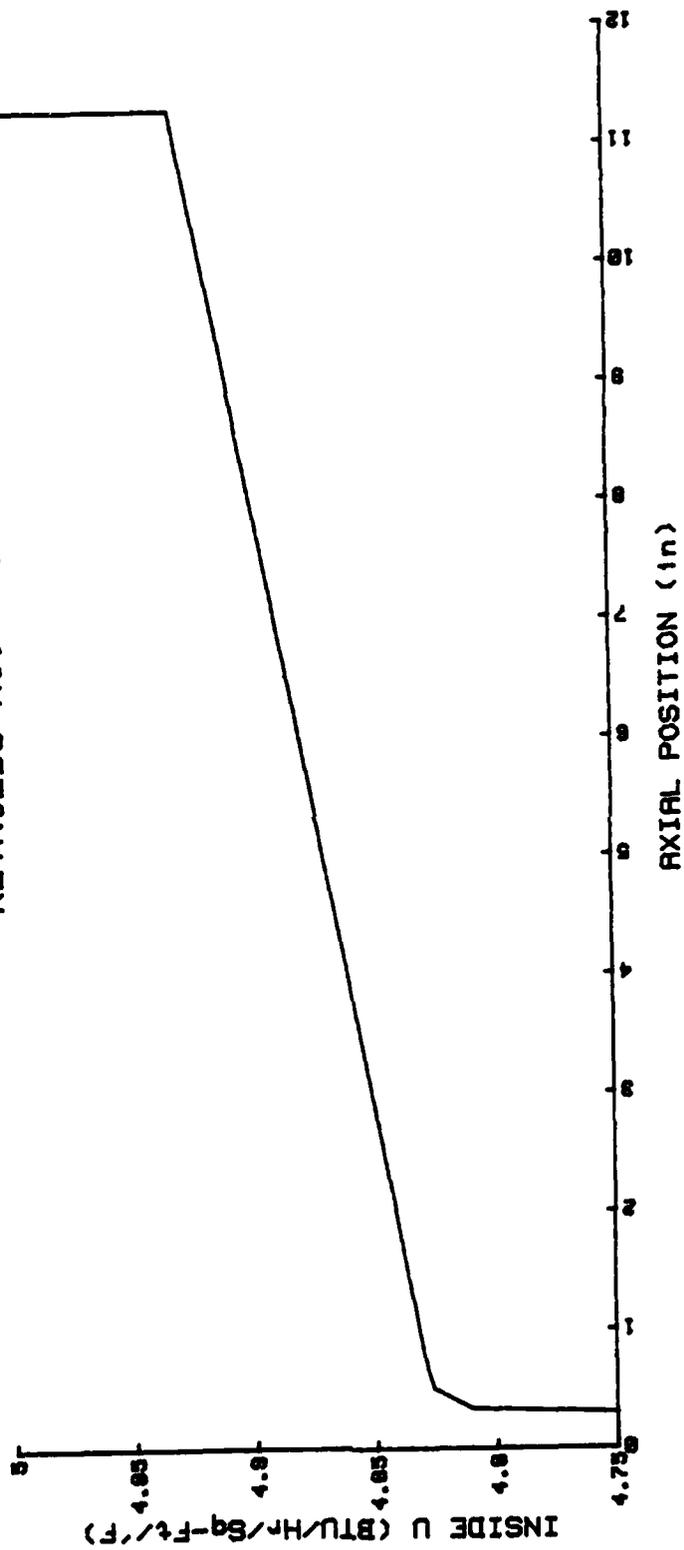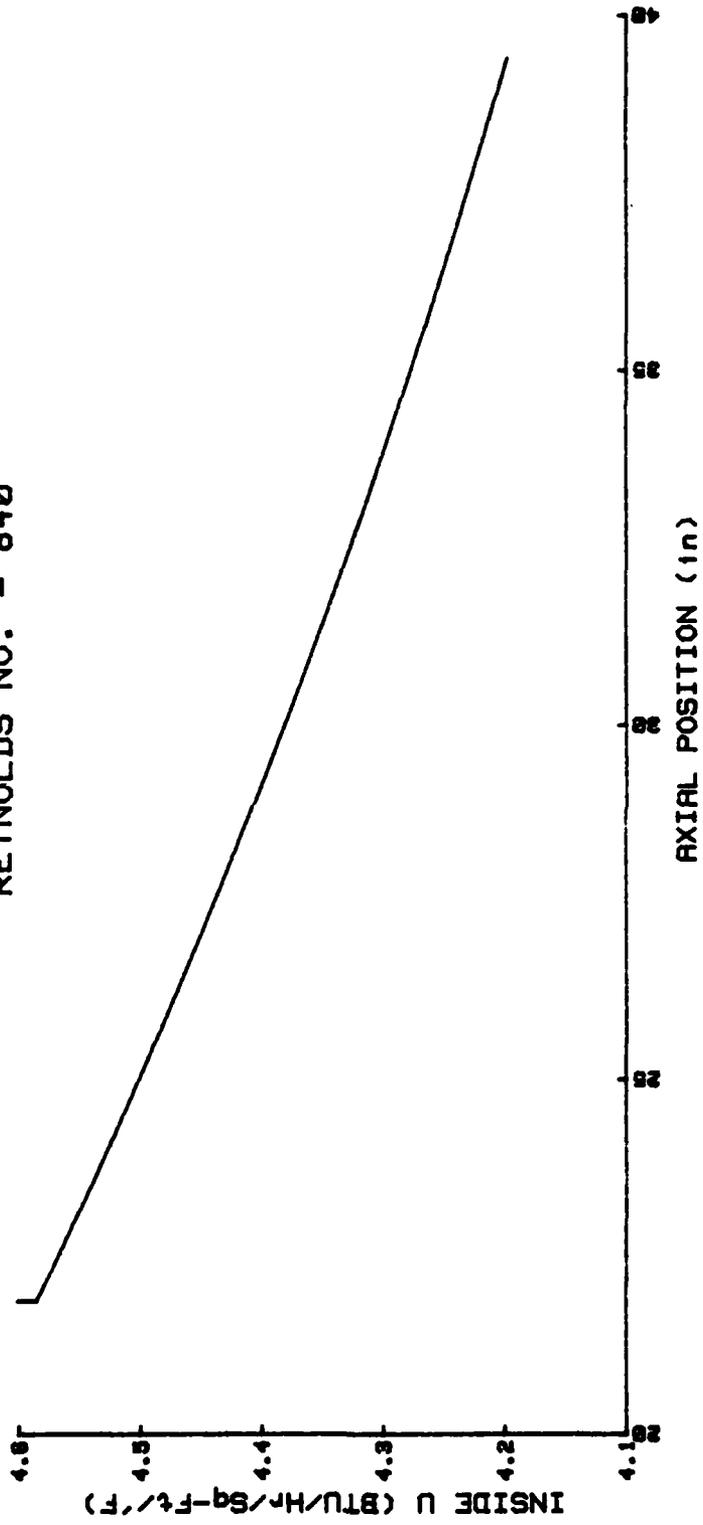
INSIDE U (BTU/Hr/Sq-ft/°F)

AXIAL POSITION (in)

FIGURE 57b

130

REYNOLDS NO. = 840

FIGURE 57c

131

REYNOLDS NO. = 840

WATER BULK TEMPERATURE (°F)

AXIAL POSITION (in)

FIGURE 58a

132

REYNOLDS NO. = 840

FIGURE 58b

REYNOLDS NO. = 840

LOCAL QUALITY (%)

AXIAL POSITION (in)

FIGURE 59a

134
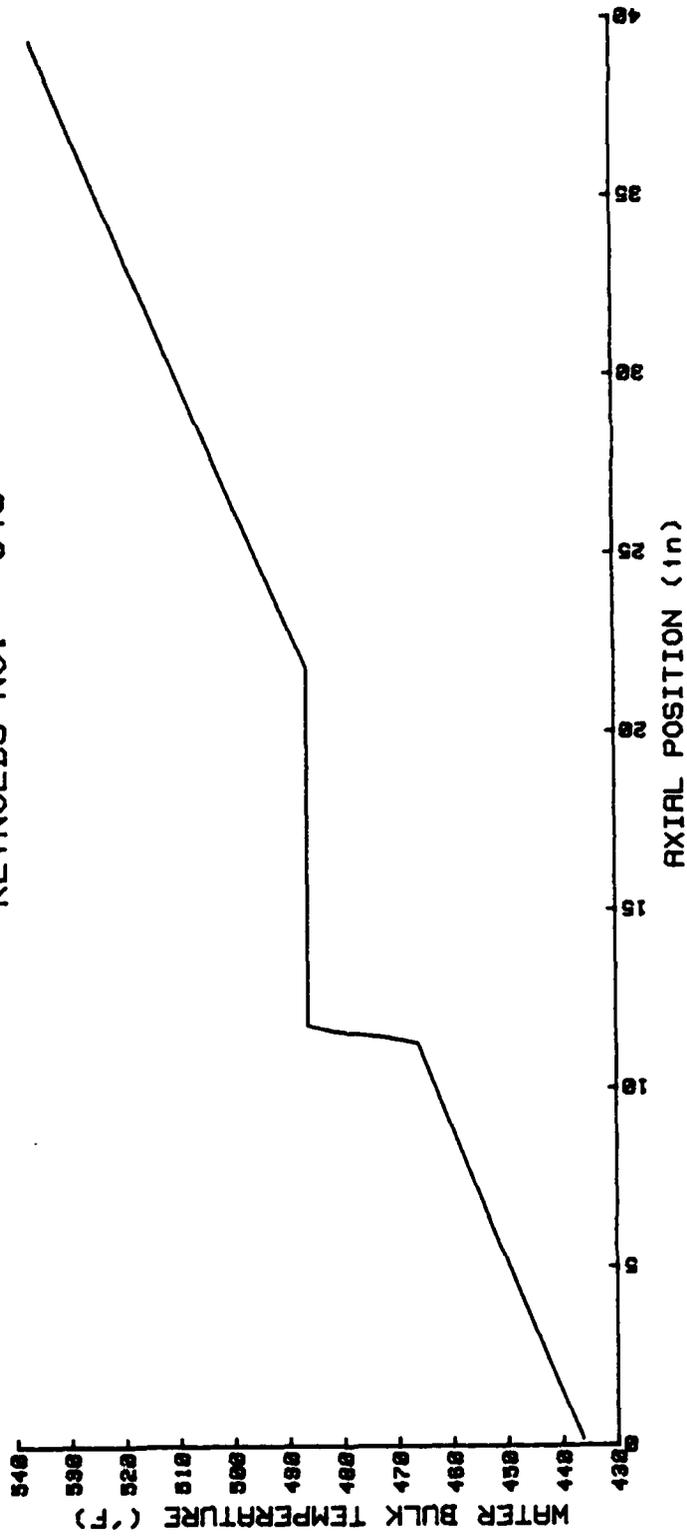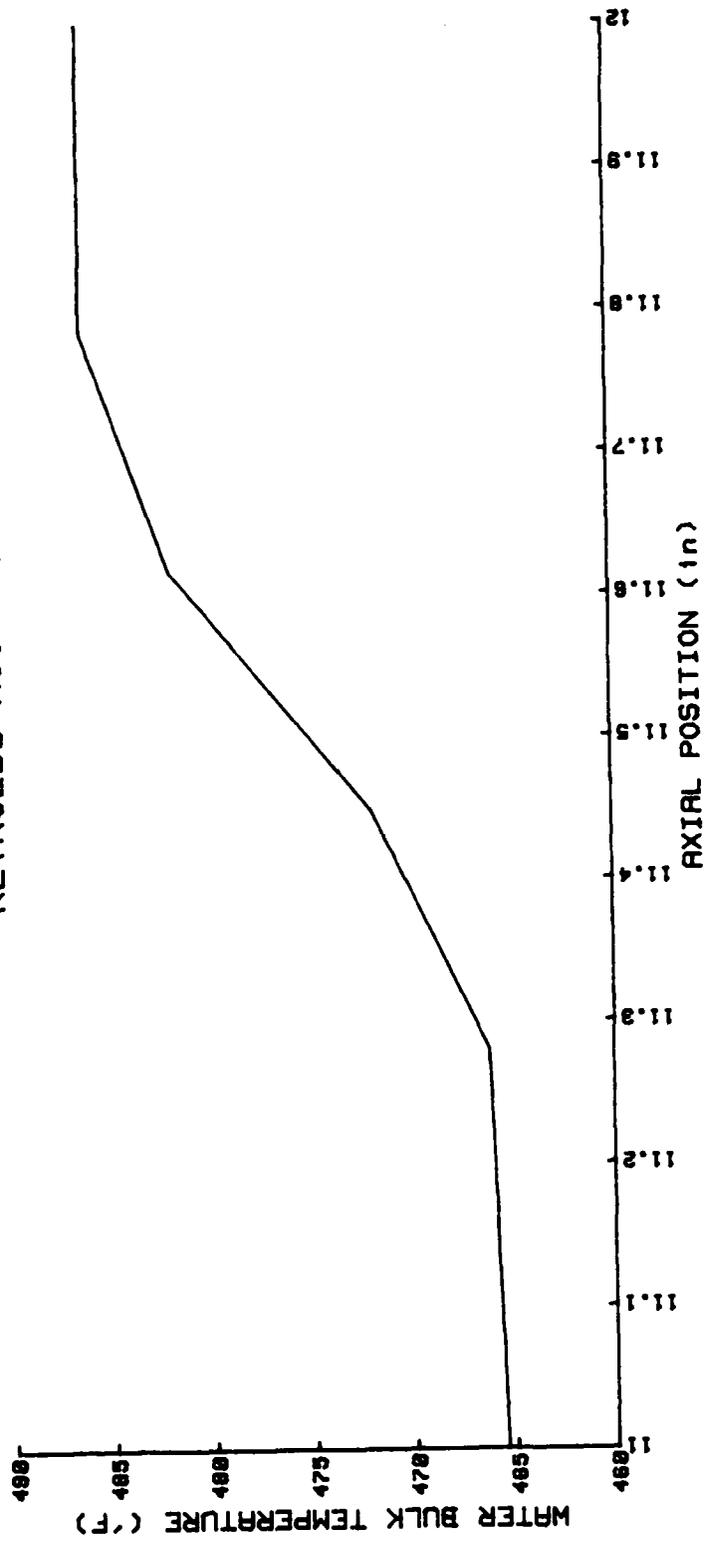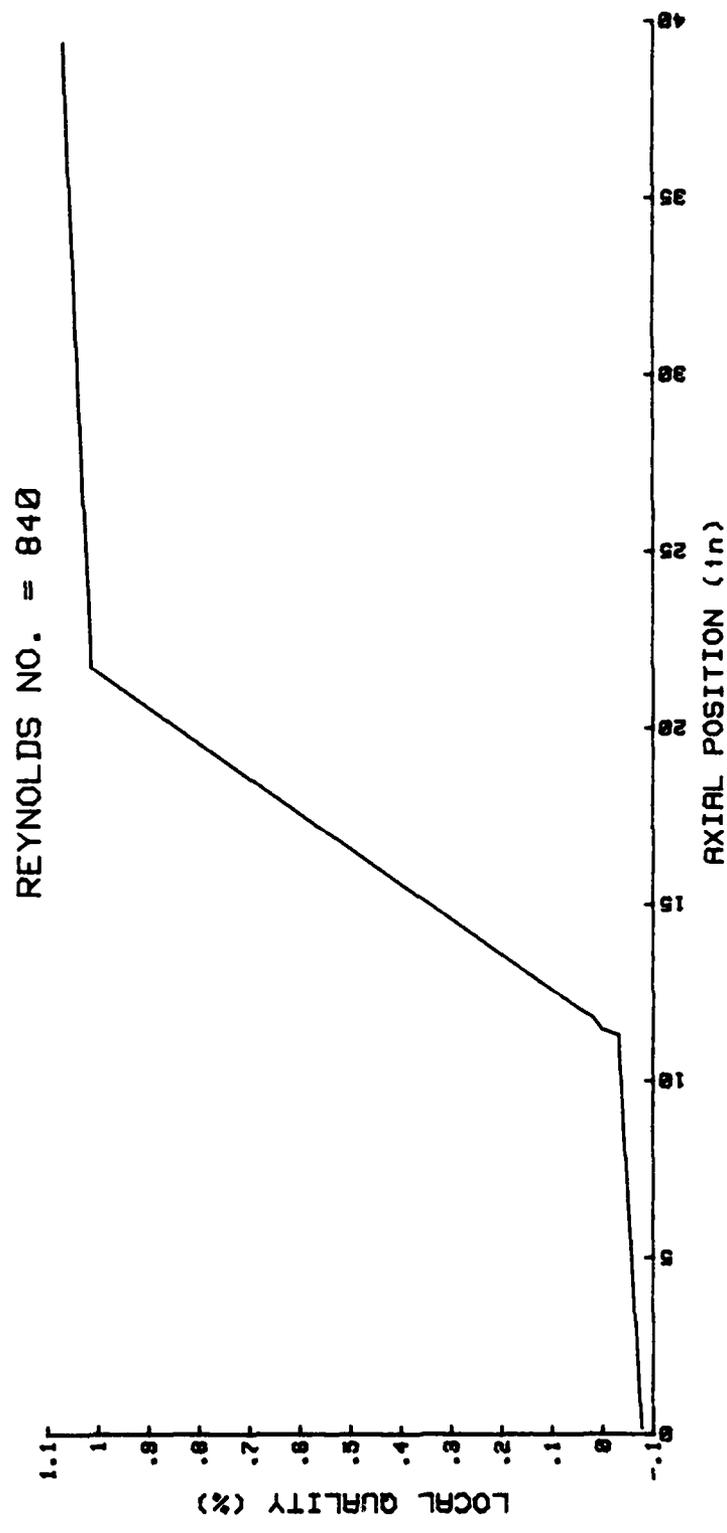
REYNOLDS NO. = 840

FIGURE 59b

135
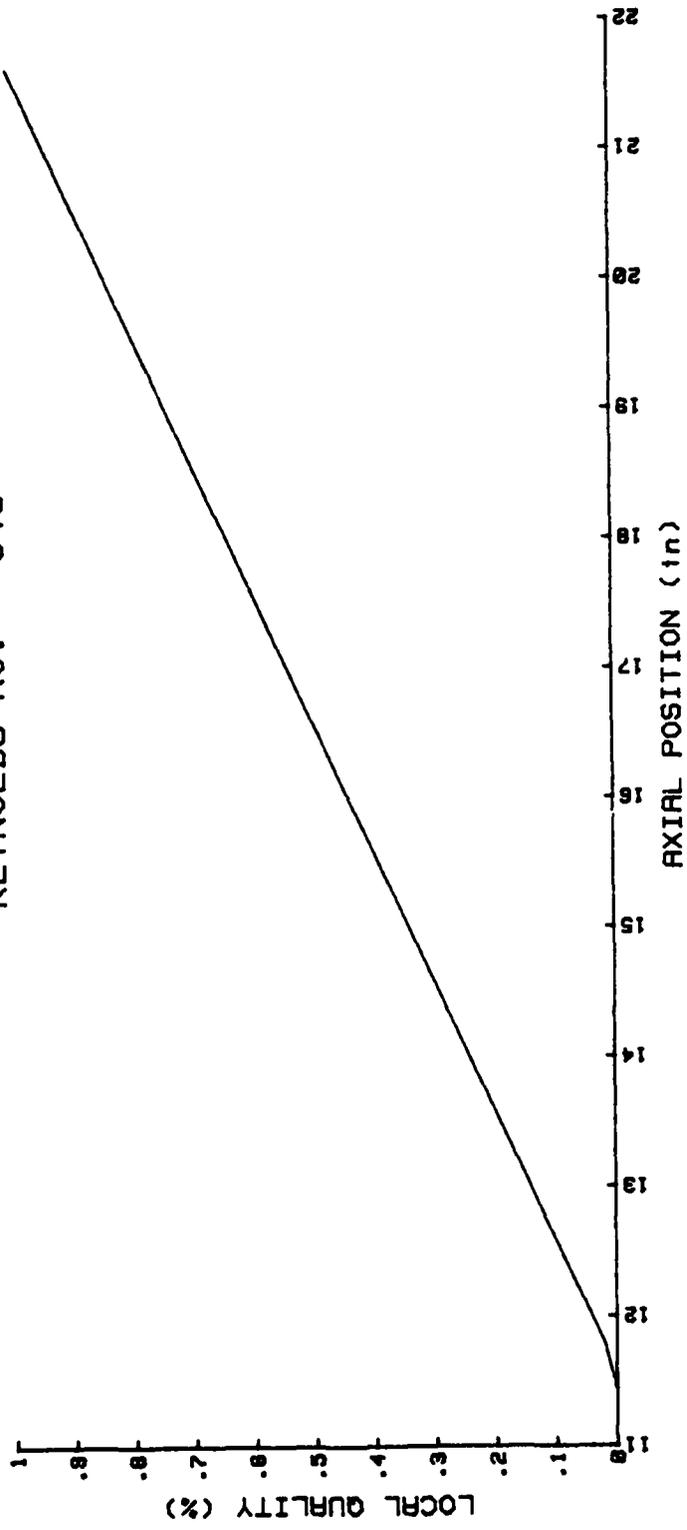
IV. <u>RECOMMENDATIONS FOR FURTHER RESEARCH</u>

Investigation of the Short Straight Tube Boiler model in
this thesis was essentially the adjustments of the water/
steam mass flow rate to determine some possible operating
conditions and heat transfer characteristics. This investi-
gation was conducted upon only one tube under steady state
steady flow operation. Additionally, the model consisted of
preheat, boiling, and superheat with accompanying single
phase flow.

In order to increase the precision of the model and lead
to greater insight into the possible performance of the S.S.T.,
consideration must be given to the inclusion of two phase
heat transfer; *in particular to* "frothy" flow correlations.
Possible sources of this information may exist in research
and experimentation conducted in "flash-type" water distilla-
tion systems.

Secondly, consideration should be given to apply enhanced
heat transfer configurations to the inside of the tubes;
helical wire inserts or helical vane inserts. This enhance-
ment would be required to increase the heat transfer in the
preheat and superheat sections and would be very important
especially as gas temperatures decreased.

Finally, the program should be transferred to the IBM
3033 computer in order to more efficiently process an expanded

program to determine actual heat exchanger size. Although a powerful tool the HP 9845-B lacks the storage capability, especially, disk storage, to handle a complete heat exchanger design analysis that would be required in order to make a reasonable comparison with other heat exchangers.

With the aforementioned analytic changes implemented, the calculated heat transfer coefficients should be compared with those coefficients obtained experimentally. The single tube experiment should also include the capability for the observation of the boiling regime in order to gain possible greater qualitative insight into the most probable flow patterns in existence in the boiling regions within the tube.

```
10 Alpha: DEF FNTanh(X)=EXP(X)/(EXP(X)+EXP(-X))*2+1
20    DEF FNRhoair(T,P)=144.00*P/(53.345*(T+459.69))
30    DEF FNCjseg(R)=.0396176083718-.0033474158863S*LOG(R)
40    DEF FNCfseg(R)=1.0882481S212-.0771218846786*LOG(R)
50    DEF FNDb(V,K,D,P,F)=.023*(K/D)*R^.8*F^.4
60    DEF FNLam(B,W,K,D,P,R,L)=1.86*(K/D)*(R*P)^(1/3)*(D/L)^(1/3)*(B/W)^.14
70    DEF FNColl(K,D,R,F,W,Rd,Gc,B,V,T)=.17*(K/D)*R^.33*(F/W)^.43*(F/W)^.25*(D^3*Rd/
2*Gc*B*T/V^2)^.10
80    DEF FNBoiltest(P,T,Q)=T+(Q/(15.6*P^1.156))/(P^.0234/2.3)
90    Gc=32.174
100   !
110   !   THIS SEGMENT OF THE PROGRAM ALLOWS THE OPERATOR TO INPUT ALL REQUIRED
VARIABLES TO PERFORM "SST" BOILER TYPE HEAT EXCHANGER ANALYSIS
120   !
130   OPTION BASE 1
140   SHORT Tg(234),Tw(234),X(234),Qa(234),Uo(234),Ui(234),Hi(234),Ho(234),Wo(23
4),Wi(234),T1(234),T2(234),Re(45),Super(45),Mdot(45),Zero(45),One(45)
150   DIM M$[2],Tube(3),Fin(9),Seg(9),Pin(7)
160   !
170   !********************************************************************************
180   !*                                                                            *
190   !*    THE FOLLOWING IS A DESCRIPTIVE LISTING OF ARRAYS:                        *
200   !*                                                                            *
210   !*    Tg .......... GAS BULK TEMPERATURE                                       *
220   !*    T1 .......... GAS ENTRY TEMPERATURE TO A ROW OF TUBES                     *
230   !*    T2 .......... GAS EXIT TEMPERATURE FROM A ROW OF TUBES                    *
240   !*    Tw .......... WATER BULK TEMPERATURE                                      *
250   !*    Wo .......... OUTSIDE TUBE-WALL TEMPERATURE                               *
260   !*    Wi .......... INSIDE TUBE-WALL TEMPERATURE                                *
270   !*    Qa .......... INSIDE SURFACE HEAT FLUX                                    *
280   !*    Hi .......... WATER-SIDE HEAT TRANSFER COEFFICIENT                        *
290   !*    Ho .......... GAS-SIDE HEAT TRANSFER COEFFICIENT                          *
300   !*    Uo .......... OVERALL HEAT TRANSFER COEFFICIENT BASED                     *
310   !*                      UPON THE OUTSIDE AREA                                   *
320   !*    Ui .......... OVERALL HEAT TRANSFER COEFFICIENT BASED                     *
330   !*                      UPON THE INSIDE AREA                                    *
340   !*                                                                            *
350   !*    X .......... STEAM QUALITY                                               *
360   !********************************************************************************
```

138

```
370  M$(1)="TUBE"
380  M$(2)="FIN"
390  !
400  !       INPUT TUBE DIMENSIONS AND PROPERTIES
410  !
420  PRINT "TUBE DIMENSIONS REQUIRED ARE IN INCHES"
430  ASSIGN #1 TO "TUBE:F8,0"
440  MAT READ #1,1;Tube
450  PRINT "1) TUBE OUTSIDE DIAMETER";TAB(50),Tube(1)
460  PRINT "2) TUBE INSIDE DIAMETER";TAB(50),Tube(2)
470  PRINT "3) TUBE LENGTH";TAB(50),Tube(3)
480  Tube:      Do=Tube(1)/12
490  Di=Tube(2)/12
500  Lt=Tube(3)/12
510  CALL Metals(1,Ktube,M$(1))
520  F$="0"
530  Fin$="SEGMENTED-FIN"
540  !
550  !       INPUT SEGMENTED FIN DIMENSIONS AND PROPERTIES
560  !
570  PRINT "ALL SEGMENTED FIN VARIABLES ARE IN INCHES "
580  ASSIGN #1 TO "SEG:F8,0"
590  MAT READ #1,1;Seg
600  MAT Fin=Seg
610  Ps=9
620  Display:      PRINT ""
630  PRINT ""
640  PRINT ""
650  IF Ps=9 THEN PRINT TAB(1),"(1)";TAB(9),"(2)";TAB(18),"(3)";TAB(27),"(4)";
660  IF Ps=9 THEN PRINT TAB(36),"(5)";TAB(45),"(6)";TAB(54),"(7)";TAB(63),"(8)";TAB(72),"(9)"
670  PRINT ""
680  IF Ps=9 THEN PRINT TAB(2),"NF";TAB(10),"NS";TAB(19),"L";TAB(28),"TF";TAB(3
7),"SN";TAB(46),"SP";TAB(55),"DF";TAB(64),"WS";TAB(73),"LC"
690  PRINT ""
700  IF Ps=9 THEN PRINT TAB(1),Fin(1);TAB(9),Fin(2);TAB(18),Fin(3);TAB(27),Fin(
4);TAB(36),Fin(5);TAB(45),Fin(6);TAB(54),Fin(7);TAB(63),Fin(8);TAB(72),Fin(9)
710  Fin:      Nf=Fin(1)
720  Nfin=INT(Lt*Nf*12+1)
```

```
730   Lt=PROUND(Nfin/Nf,-3)
740   Lt=Lt/12
750   Ns=Fin(2)
760   L=Fin(3)/12
770   Tf=Fin(4)/12
780   Sn=Fin(5)/12
790   Sp=Fin(6)/12
800   Df=Fin(7)/12
810   Ws=Fin(8)/12
820   Lc=Fin(9)/12
830   PRINTER IS 0
840   PRINT ""
850   PRINT ""
860   PRINT "THE SELECTED FIN PROFILE IS THE";TAB(33),Fin$
870   PRINT ""
880   Abto=PROUND(PI*Do*Lt,-5)
890   Abti=PROUND(PI*Di*Lt,-5)
900   Dr=Do
910   PRINT "THE OUTSIDE TUBE DIAMETER IS";TAB(43),Tube(1);TAB(55),"INCHES"
920   PRINT "THE INSIDE TUBE DIAMETER IS";TAB(43),Tube(2);TAB(55),"INCHES"
930   PRINT "THE TUBE LENGTH IS";TAB(42),Tube(3);TAB(55),"INCHES"
940   PRINT "THE TOTAL OUTSIDE TUBE AREA IS";TAB(43),Abto;TAB(55),"SQUARE FEET"
950   PRINT "THE TOTAL INSIDE TUBE IS";TAB(43),Abti;TAB(55),"SQUARE FEET"
960   PRINT "THE FIN ROOT DIAMETER IS";TAB(43),Tube(1);TAB(55), "INCHES"
970   PRINT "THE NUMBER OF 'FINS' PER INCH IS";TAB(43),Fin(1)
980   PRINT "THE NUMBER OF 'SEGMENTS' PER FIN IS";TAB(43),Fin(2)
990   PRINT "THE FIN HEIGHT IS";TAB(43),Fir(3);TAB(55),"INCHES"
1000  PRINT "THE FIN THICKNESS IS";TAB(44),Fin(4);TAB(55),"INCHES"
1010  PRINT "THE TRANSVERSE TUBE PITCH IS";TAB(43),Fin(5),"INCHES"
1020  PRINT "THE LONGITUDINAL TUBE PITCH IS";TAB(43),Fin(6);TAB(55),"INCHES"
1030  PRINT "THE FIN OUTSIDE DIAMETER IS";TAB(43),Fin(7);TAB(55),"INCHES"
1040  PRINT "THE FIN SEGMENT WIDTH IS";TAB(44),Fin(8);TAB(55),"INCHES"
1050  PRINT "THE LENGTH OF CUT FROM FIN TIF IS";TAB(44),Fin(9);TAB(55),"INCHES"
1060  PRINT "THE TOTAL INTEGER NUMBER OF FINS IS";TAB(41),Nfin;TAB(55),"FINS"
1070  PRINT "ACTUAL TUBE LENGTH WILL BE";TFB(42),Lt*12;TAB(55),"INCHES"
1080  PRINTER IS 16
1090  Ntrmax=12/Sn
1100  Ntr=Ntrmax
1110  PRINT "THE MAXIMUN NUMBER OF TUBES ALLOWED IN ANY ROW IS",Ntrmax
```

```
1120    CALL Metals(2,Kfin,M$(2))
1130    PRINTER IS 0
1140    PRINT "THE FIN THERMAL CONDUCTIVITY IS";TAB(41),Kfin;TAB(55),"BTU/Hr.Ft./F
"
1150    PRINT "THE TUBE THERMAL CONDUCTIVITY IS";TAB(42),Ktube;TAB(55),"BTU/Hr.Ft.
/F"
1160    PRINT ""
1170    !
1180    !        CALCULATE HEAT TRANSFER AREAS
1190    !
1200    Afint=PROUND(Nfin>(Ns*(2*Lc*Ns+2*Tf*Lc+Tf*Ns)+PI/2*((Df-2*Lc)^2-Do^2)),-6)
1210    Abt=PROUND(Abto-Nfin*PI*Do*Tf,-6)
1220    Afin=PROUND(Afint/Nfin,-6)
1230    Ato=PROUND(Afint+Abt,-6)
1240    Aso=PROUND(Ato/Nfin,-6)
1250    Asi=PROUND(Abti/Nfin,-6)
1260    Afhe=PROUND(Lt*Sn*Ntr,-6)
1270    Af=PROUND(Afhe/(Ntr*Nfin),-6)
1280    PRINT ""
1290    PRINT "CALCULATED HEAT TRANSFER AREAS IN SQUARE FEET:"
1300    PRINT ""
1310    PRINT TAB(10),"TOTAL HEAT EXCHANGER FRONTAL AREA";TAB(50),Afhe
1320    PRINT TAB(10),"TOTAL FIN AREA IS";TAB(50),Afint
1330    PRINT TAB(10),"TOTAL BARE TUBE AREA IS";TAB(51),Abt
1340    PRINT TAB(10),"TOTAL OUTSIDE TUBE AREA IS";TAB(50),Ato
1350    PRINT TAB(10),"SINGLE FIN AREA IS";TFB(52),Afin
1360    PRINT TAB(10),"OUTSIDE 'ELEMENTAL' AREA IS";TAB(52),Aso
1370    PRINT TAB(10),"INSIDE 'ELEMENTAL' AREA IS";TAB(52),Asi
1380    PRINT TAB(10),"ELEMENTAL' FRONTAL AREA IS";TAB(52),Af
1390    Ab=PROUND((Do*Lt+2*Nfin*L*Tf)/Nfin,-6)
1400    Amin=PROUND(Af-Ab,-6)
1410    PRINT TAB(10),"ELEMENTAL' BLOCKED AREA IS";TAB(52),Ab
1420    PRINT TAB(10),"ELEMENTAL' MINIMUM GFS-FLOW AREA IS";TAB(52),Amin
1430    PRINT ""
1440    PRINT ""
1450    PRINTER IS 16
1460 Agibt: INPUT "ENTER AVERAGE GAS INLET BULK TEMPERATURE",Spec(1)
1470 Gsmfr: INPUT "ENTER GAS-SIDE MASS FLOW RATE ( LBM/SEC )",Spec(2)
1480 Gsp:   INPUT "ENTER GAS-SIDE PRESSURE ( PSIA )",Spec(3)
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1490 Wsp: INPUT "ENTER WATER-SIDE PRESSURE ( PSIA )",Spec(4)
1500 Tsc=0
1510 PRINT "POSSIBLE WATER INLET CONDITIONS"
1520 PRINT "    0=SUBCOOLED"
1530 PRINT "    1=SATURATED"
1540 INPUT "ENTER INLET CONDITION",Inlet$
1550 IF Inlet$="0" THEN INPUT "ENTER AMOUNT OF INLET SUBCOOLING ('F)",Spec(6)
1560 !
1570 !    GIVEN AN EFFECTIVENESS, E, THE CHANGE IN FLUID BULK TEMPERATURES CAN
BE FOUND
1580 !
1590 INPUT "ENTER AN ASSUMED INITIAL HEAT EXCHANGER EFFECTIVENESS",Spec(7)
1600 PRINT "DUE TO DISK STORAGE LIMITATIONS THE MAXIMUM NUMBER OF DIFFERENT REY
NOLDS WHICH CAN BE INVESTIGATED AT ONE TIME IS 45"
1610 INPUT "HOW MANY DIFFERENT REYNOLDS ARE TO BE INVESTIGATED",Reyno
1620 FOR Run=1 TO Reyno
1630 INPUT "ENTER THE VALUE FOR THE REYNOLDS NUMBER",Spec(5)
1640 IF Run>1 THEN Cal
1650 Listing:   PRINT "VALUES OF THE SPECIFIED 'DESIGN' PARAMETERS:"
1660 PRINT ""
1670 PRINT TAB(10),"1) GAS 'NLET TEMPERATURE TO HEAT EXCHANGER IS";TAB(55),Spec
(1);TAB(60),"'F."
1680 Tsat=PROUND(FNSattemp((Spec(4))),-2)
1690 PRINT TAB(10),"2) GAS-SIDE TOTAL MASS FLOWRATE IS";TAB(44),Spec(2);TAB(49)
,"Lbm/Sec."
1700 PRINT TAB(10),"3) GAS-SIDE PRESSURE IS";TAB(33),Spec(3);TAB(41),"psia."
1710 PRINT TAB(10),"4) WATER-SIDE PRESSURE IS";TAB(35),Spec(4);TAB(40),"psia."
1720 PRINT TAB(10),"    THE CORRESPONDING SATURATION TEMPERATURE IS";TAB(56),PRO
UND(Tsat,-2);TAB(64),"'F."
1730 IF Inlet$="0" THEN PRINT TAB(10),"5) THE WATER ENTERS SUBCOOLED AN AMOUNT
OF ";TAB(53),Spec(6);TAB(57),"'F."
1740 IF Inlet$="1" THEN PRINT TAB(10),"5) THE WATER ENTERS AT THE SATURATION TE
MPERATURE"
1750 PRINT TAB(10),"6) THE ASSUMED HEAT EXCHANGER EFFECTIVENESS IS";TAB(56),Spe
c(7);TAB(61),"."
1760 IF Count>0 THEN PRINT ""
1770 IF Count>0 THEN PRINT ""
1780 IF Count>0 THEN Cal
1790 Cor$="NULL"
```

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```
1800  Hardcopy:  Count=1
1810  PRINTER IS 0
1820  GOTO Listing
1830  Cal:  PRINTER IS 16
1840  Tgi=Spec(1)
1850  Tgin=Tgi
1860  Tgf=Tgi
1870  Tgb=Tgi
1880  MAT T1=(Tgi)
1890  Mfrgin=3600*Spec(2)
1900  Mdotg=Mfrgin/(Ntr*Nfin)
1910  Cpg=FNAircp((Tgf))
1920  Pg=Spec(3)
1930  Pw=Spec(4)
1940  CALL Combstm((Pw),(Tsat),Istm,Sstm,Rfostm)
1950  Iw=FNHsl((Tsat))
1960  Ifg=Istm-Iw
1970  Tsc=Spec(6)
1980  Twi=PROUND(Tsat-Tsc,-2)
1990  IF Inlet$="0" THEN CALL Entcw((Pw),(Twi),Iwi)
2000  IF Inlet$="1" THEN Iwi=FNHsl((Tsat))
2010  Iwin=Iwi
2020  Iwout=Iwi
2030  Twin=Twi
2040  Twout=Twin
2050  Diff=Tsat-Twout
2060  Tub=Twi
2070  IF Inlet$="0" THEN CALL Viscw((Pw),(Tub),Vw)
2080  IF Inlet$="1" THEN Vw=FNSatwvisc((Tsat))
2090  Rewin=Spec(5)
2100  Mdotw=PI*Di*Rewin*Vw/4
2110  Rew=Rewin
2120  Caprtg=Mdotgo*FNAircp((Tgf))
2130  IF Inlet$="0" THEN CALL Sphcw((Pw),(Tub),Cpw)
2140  IF Inlet$="0" THEN Caprtw=Mdotw*Cpw
2150  IF Inlet$="1" THEN Caprtw=Mdotw*FNSprsatw((Tsat))
2160  Tto=Tgin-1
2170  Tti=Tto-1.5
2180  !
```

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```
2190  !        MAIN PROGRAM
2200  !
2210  Iteration: FOR Axial=1 TO Nfin
2220  PRINT "SEGMENT";TAB(15.00),Axial
2230      Tgin=Ti(Axial)
2240      Dtmax=PROUND(Tgin-Twin,-2)
2250  Effectiveness: Eo=Eff
2260      Caprtg=Mdotg*Cpg
2270  Cminandouttemp:   CALL Caprate(Twin,Twt,Twout,(Pw),(Xt),(Tsat),(Mdotw),Caprt
w,Type$)
2280      IF Type$="SUPERHEAT" THEN B$="NULL"
2290      IF Type$<>"SATURATED" THEN Mincapacity
2300      C$="CMAX-UNMIXED"
2310      Cmin=Caprtg
2320      Qs=Cmin*Dtmax*Eo
2330      Tgout=PROUND(Tgin-Qs/Caprtg,-2)
2340      GOTO Quality
2350  Mincapacity: Cmin=MIN(Caprtg,Caprtw)
2360      Cmax=MAX(Caprtg,Caprtw)
2370      IF Cmax=Caprtg THEN C$="CMAX-MIXED"
2380      IF Cmax=Caprtw THEN C$="CMAX-UNMIXED"
2390      Qs=Cmin*Dtmax*Eo
2400      Tgout=PROUND(Tgin-Qs/Caprtg,-2)
2410  Quality: Iwout=Iwin+Qs/Mdotw
2420      Iw=FNHsl((Tsat))
2430  CALL Combstm((Pw),(Tsat),Ismt,Sstm,Rhostm)
2440      Ifg=Istm-Iw
2450      IF Tsat-Twb>=20.00 THEN True
2460      IF Twb>=Tsat THEN True
2470      Cpw=FNSphsatw((Tsat))
2480      Xd=PROUND(-Cpw*(Tsat-Twb)/Ifg,-4)
2490      Xe=PROUND((Iwout-Iw)/Ifg,-4)
2500      Xt=PROUND(Xe-Xd*EXP(Xe/Xd-1),-4)
2510      Qv=Qs*(1-EXP(Xt/Xd-1))
2520      Qc=Qs-Qv
2530      GOTO Bulktemps.
2540  True: Xt=PROUND((Iwout-Iw)/Ifg,-4)
2550  Bulktemps: IF Type$="SUPERHEAT" THEN Twout=Twin+Qs/Caprtw
2560      IF (Type$="SUBCOOLED") AND (A$="SAT") THEN Twout=Twin+Qc/Caprtw
```

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```
2570 IF (Type$="SUBCOOLED") AND (Twout>Tsat) THEN Twout=Tsat
2580 Twb=PROUND((Twout+Twin)/2,-2)
2590 Tgb=PROUND((Tgout+Tgin)/2,-2)
2600 Twf=PROUND((Twb+Tti)/2,-2)
2610 Tgf=PROUND((Tgb+Tto)/2,-2)
2620 Airproperties:   Cpg=FNAircp((Tgf))
2630 Kg=FNAirtc((Tgf))
2640 Prg=FNAirprandtl((Tgf))
2650 Visg=FNAirvisc((Tgf))
2660 Rhog=FNRhoair((Tgf),(Pg))
2670 Prgw=FNAirprandtl((Tto))
2680 Waterproperties: IF (Type$="SATURATED") OR (B$="BOILING") THEN Saturation
2690 IF (Xt>=1) AND (Rew>2200) THEN Stmdb
2700 IF (Xt>=1) AND (Rew>=2000) AND (Rew<=2200) THEN Stmlam
2710 IF (Xt>=1) AND (Rew<2000) THEN Stmcoll
2720 IF Rew>2200 THEN Dittusboelter
2730 IF (Rew>=2000) AND (Rew<=2200) THEN Laminar
2740 IF Rew<2000 THEN Collier
2750 Saturation: A$="SAT"
2760 Rvw=FNSatwvisc((Tsat))
2770 Rrhow=1/FNVolcw((Pw),(Tsat))
2780 CALL Combstm((Pw),(Tsat),Istm,Sstm,Rhostm)
2790 Rcpw=FNSphsatw((Tsat))
2800 Rprw=FNSatwprno((Tsat))
2810 Iw=FNHsl((Tsat))
2820 Ifg=Istm-Iw
2830 Sigma=5.8E-3*(1-.000142*Tsat)
2840 IF Rew>10000 THEN Satst
2850 IF (Rew>2200) AND (Rew<=10000) THEN Satdb
2860 IF (Rew>=2000) AND (Rew<=2200) THEN Satlam
2870 IF Rew<2000 THEN Satcoll
2880 Dittusboelter: A$="D-B"
2890 CALL Viscw((Pw),(Twb),Vwb)
2900 CALL Tccw((Pw),(Twb),Kw)
2910 CALL Sphcw((Pw),(Twb),Cpw)
2920 GOTO Dbpr
2930 Satdb: Vwb=FNSatwvisc((Tsat))
2940 Kw=FNSatwtc((Tsat))
2950 Cpw=FNSphsatw((Tsat))
```

```
2960      GOTO Dbpr
2970 Stmdb: A$="D-B:STM"
2980      Vwb=FNViscs((Pw),(Twb))
2990      CALL Sphs((Pw),(Twb),Cpw)
3000      CALL Tcs((Pw),(Twb),Kw)
3010 Dbpr: Prwb=Vwb*Cpw/Kw
3020      Rew=4*Mdotw/(PI*Di*Vwb)
3030      IF Rew<2200 THEN Waterproperties
3040      GOTO Airfactors
3050 Laminar: A$="LAM"
3060      CALL Viscw((Pw),(Twb),Vwb)
3070      CALL Tccw((Pw),(Twb),Kw)
3080      CALL Sphcw((Pw),(Twb),Cpw)
3090      CALL Viscw((Pw),(Tti),Vww)
3100      GOTO Lampr
3110 Satlam: Vwb=FNSatwvisc((Tsat))
3120      Kw=FNSatwtc((Tsat))
3130      Cpw=FNSphsatw((Tsat))
3140      Vww=FNSatwvisc((Tti))
3150      GOTO Lampr
3160 Stmlam: A$="LAM:STM"
3170      Vwb=FNViscs((Pw),(Twb))
3180      Vww=FNViscs((Pw),(Twb))
3190      CALL Sphs((Pw),(Twb),Cpw)
3200      CALL Tcs((Pw),(Twb),Kw)
3210 Lampr: Prwb=Vwb*Cpw/Kw
3220      Rew=4*Mdotw/(PI*Di*Vwb)
3230      IF Rew>2200 THEN Waterproperties
3240      GOTO Airfactors
3250 Collier: A$="COLL"
3260      CALL Viscw((Pw),(Twf),Vwf)
3270      CALL Viscw((Pw),(Tti),Vww)
3280      CALL Tccw((Pw),(Twf),Kwf)
3290      CALL Tccw((Pw),(Tti),Kww)
3300      CALL Sphcw((Pw),(Twf),Cpwf)
3310      CALL Sphcw((Pw),(Tti),Cpww)
3320      GOTO Dummy
3330 Satcoll: Kwf=FNSatwtc((Twf))
3340      Kww=FNSatwtc((Tti))
```

```
3350        Cpwf=FNSphsatw((Twf))
3360        Cpww=FNSphsatw((Tti))
3370        Vww=FNSatwvisc((Tti))
3380        Vwf=FNSatwvisc((Twf))
3390        GOTO Dummy
3400 Stmcol1: A$="COLL:STM"
3410        CALL Tcs((Pw),(Twf),Kwf)
3420        CALL Tcs((Pw),(Tti),Kww)
3430        CALL Sphs((Pw),(Twf),Cpwf)
3440        CALL Sphs((Pw),(Tti),Cpww)
3450        Vww=FNViscs((Pw),(Tti))
3460        Vwf=FNViscs((Pw),(Twf))
3470 Dummy: Dummy=FNVolcw((Pw),Twf+1)
3480        Rhowf=1/FNVolcw((Pw),(Twf))
3490        Beta=LOG(Rhowf*Dummy)
3500        Prww=Vww*Cpww/Kww
3510        Prwf=Vwf*Cpwf/Kwf
3520        Rew=4*Mdotw/(PI*Di*Vwf)
3530        IF Rew>=2000 THEN Waterproperties
3540 Airfactors: Reg=Mdotg/Amin*Do/Visg
3550        J=FNCjseg((Reg))
3560        F=FNCfseg((Reg))
3570 Airheatxfercoef:   Hout=Kg*J*Reg*Prg^(1/3)/Do
3580 H2Oheatxfercoef:   IF Rew>2200 THEN Hc=FNDb(Vwb,Kw,Di,Prwb,Rew)
3590        IF (Rew>=2000: AND (Rew<=2200) THEN Hc=FNLam(Vwb,Vww,Kw,Di,Prwb,Rew,Lt
)
3600        Dt=Tti-Twb
3610        IF Rew<2000 THEN Hc=FNColl((Kwf,Di,Rew,Prwf,Prww,Rhowf,Gc,Beta,Vwf,Dt)
3620        IF A$<>"SAT" THEN Hin=Hc
3630        IF A$<>"SAT" THEN Finefficiency
3640        Hb=Rww*SQR((Rrhow-Rhostm)/Sigma)*(Rcpw/(.006*Rprw))^3*((Tti-Tsat)/lfg)
^2
3650        Q=Hc*(Tti-Twb)+Hb*(Tti-Tsat)
3660        Hin=Q/(Tti-Twb)
3670 Finefficiency: M1=(Df-Do)/2*SQR(2*(Ws+Tf)*Ifg/(Ws*Tf*Kfin))
3680        Etaf=FNTanh((M1))/M1
3690 Surfaceeffect:   Etas=1-(1-Etaf)*Afin/Fso
3700 Overalhtxfercof: Uoo=1/(Aso/Asi*1/Hin+Aso*LOG((Do/Di)/(2*PI*Ktube*Lt/Nfin)+1
/(Hout*Etas))
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3710    Uoi=1/(1/Hin+Asi*LOG(Do/Di)/(2*PI*Ktube*Lt/Nfin)+Asi/Aso*1/(Hout*Etas)
)
3720 Numberxferunits:  Ntu=.5*1/Cmin*(Uoo*Fso+Uoi*Asi)
3730 Segefectiveness:  IF Type$="SATURATED" THEN Phasechange
3740    C=Cmin/Cmax
3750    IF C$="CMAX-MIXED" THEN Eff=1/C*(1-EXP(-C*(1-EXP(-Ntu))))
3760    IF C$="CMAX-UNMIXED" THEN Eff=1-EXP(-1/C*(1-EXP(-C*Ntu)))
3770    GOTO Resistance
3780 Phasechange: Eff=1-EXP(-Ntu)
3790 Resistance:    Ro=1/(Etas*Hout*Aso)
3800 Ri=1/(Hin*Asi)
3810 Rth=1/(Uoi*Asi)
3820 Rtube=LOG(Do/Di)/(2*PI*Ktube*Lt/Nfin)
3830 Qai=Qs/Asi
3840 Walltemps:  Tt=FNBoiltest((Pw),(Tsat),(Qai))
3850    IF (Tti>=Tt) AND ((Tsat-Twb)>=0) FND (Tsat-Twb<20.00)) THEN B$="BOILING
"
3860    IF (Tsat-Twb>=0) AND (Tsat-Twb<20.00) AND (A$<>"SAT") THEN Waterproper
ties
3870    IF A$<>"SAT" THEN Tto=Tgb-Ro/Rth*(Tgb-Twb)
3880    IF A$<>"SAT" THEN Tti=Twb+Ri/Rth*(Tgb-Twb)
3890    IF A$="SAT" THEN Tti=FNBoiltest((Pw),(Tsat),Qai)
3900    IF A$="SAT" THEN Tto=Tti+Rtube/Rth*(Tgb-Twb)
3910 Convergencetest: IF Type$="SUBCOOLED" THEN Delta=.00005
3920    IF (Type$="SUBCOOLED") AND (A$="SAT") THEN Delta=.00005
3930    IF Type$="SATURATED" THEN Delta=.000001
3940    IF Type$="SUPERHEAT" THEN Delta=.0000625
3950    IF ABS(Eo-Eff)>Delta THEN Effectiveness
3960    Qtotal=Qtotal+Qs
3970    Twin=Twout
3980    Iwin=Iwout
3990    T2(Axial)=PROUND(Tgout,-2)
4000    Tw(Axial)=PROUND(Twb,-2)
4010    X(Axial)=PROUND(Xt,-4)
4020    Uo(Axial)=PROUND(Uoo,-4)
4030    Ui(Axial)=PROUND(Uoi,-4)
4040    Ho(Axial)=PROUND(Hout,-4)
4050    Hi(Axial)=PROUND(Hin,-4)
4060    Qa(Axial)=PROUND(Qs/Asi,-4)
```

```
4070    Wo(Axial)=PROUND(Tto,-2)
4080    Wi(Axial)=PROUND(Tti,-2)
4090    Nextsegment: NEXT Axial
4100    Overalheatbal: IF Inlet$="1" THEN Vis
4110      CALL Viscw((Pw),(Twi),Vw)
4120      CALL Entcw((Pw),(Twi),Iw)
4130      GOTO Calling
4140    Vis: Vw=FNSatwvisc((Tsat))
4150      Iw=FNHsl((Tsat))
4160    Calling: IF Xt>1 THEN CALL Combstm((Pw),(Twout),Istm,Satm,Rhostm)
4170      IF Xt=1 THEN CALL Combstm((Pw),(Tsat),Istm,Satm,Rhostm)
4180      IF Xt<1 THEN Jstm=Iwout
4190      Mdotwb=Qtotal/(Istm-Iw)
4200      Check=ABS((Mdotwb-Mdotw)/Mdotwb)
4210      IF Check<.05 THEN Printout
4220      Mdotw=.5*(Mdotw+Mdotwb)
4230      Rewin=4*Mdotw/(PI*Di*Vw)
4240      Twin=Twi
4250      Iwin=Iwi
4260      Twout=Twi
4270      Twb=Twi
4280      Xt=-1
4290      Qtotal=0
4300      GOTO Iteration
4310    Printout: Re(Run)=Rewin
4320      PRINTER IS 0
4330      PRINT ""
4340      PRINT ""
4350      PRINT "RECORD NUMBER IS",Run
4360      PRINT "------------------"
4370      PRINT ""
4380      PRINT ""
4390      PRINT "ASSUMED INLET WATER-SIDE REYNOLDS NUMBER",PROUND(Spec(5),-3)
4400      PRINT ""
4410      PRINT "ACTUAL INLET WATER-SIDE REYNOLDS NUMBER",PROUND(Rewin,-3)
4420      PRINT ""
4430      PRINT "OUTLET WATER-SIDE REYNOLDS NUMBER",PROUND(Rew,-3)
4440      Re(Run)=Rewin
4450      PRINT ""
```

```
4460  PRINT "OVERALL WATER MASS FLOW RATE=",PROUND(Mdotwb,-6)
4470  Mdot(Run)=Mdotwb
4480  PRINT ""
4490  PRINT "TOTAL OVERALL HEAT ADDITION RATE=",PROUND(Qtotal,-3)
4500  PRINT ""
4510  PRINT "THE INLET WATER TEMPERATURE IS",Twi
4520  PRINT ""
4530  PRINT "THE SATURATION TEMPERATURE IS",PROUND(Tsat,-2)
4540  PRINT ""
4550  IF X(Nfin)<1 THEN PRINT "THE OUTLET WATER TEMPERATURE IS",PROUND(Twout
      ,-2)
4560  IF X(Nfin)>=1 THEN PRINT "THE OUTLET STEAM TEMPERATURE IS ",PROUND(Two
ut,-2)
4570  PRINT ""
4580  PRINT "THE AMOUNT OF SUPERHEAT IS",PROUND(Twout-Tsat,-2)
4590  Super(Run)=PROUND(Twout-Tsat,-2)
4600  PRINT ""
4610  PRINT ""
4620  PRINTER IS 16
4630  Recordresults: PRINT "RECORDING RESULTS ON DISK"
4640  FOR I=1 TO Nfin
4650  Tg(I)=PROUND(.5*(T1(I)+T2(I)),-2)
4660  NEXT I
4670  ASSIGN #1 TO "TG:F8,1"
4680  ASSIGN #2 TO "TW:F8,1"
4690  ASSIGN #3 TO "WO:F8,1"
4700  ASSIGN #4 TO "WI:F8,1"
4710  ASSIGN #5 TO "QA:F8,1"
4720  ASSIGN #6 TO "HI:F8,1"
4730  ASSIGN #7 TO "HO:F8,1"
4740  ASSIGN #8 TO "UO:F8,1"
4750  ASSIGN #9 TO "UI:F8,1"
4760  ASSIGN #10 TO "X:F8,1"
4770  MAT PRINT #1,Run;Tg
4780  MAT PRINT #2,Run;Tw
4790  MAT PRINT #3,Run;Wo
4800  MAT PRINT #4,Run;Wi
4810  MAT PRINT #5,Run;Qa
4820  MAT PRINT #6,Run;Hi
```

```
4830 MAT PRINT #7,Run;Ho
4840 MAT PRINT #8,Run;Uo
4850 MAT PRINT #9,Run;Ui
4860 MAT PRINT #10,Run;X
4870 Mdotw=0
4880 Qtotal=0
4890 FOR I=1 TO Nfin
4900 IF I=Nfin THEN 4970
4910 IF (SGN(X(I))<0) AND SGN(X(I+1))>0) THEN Loczer=Lt*12/Nfin*(I+(0-X(I))/(X(I+1)-X(I)))
4920 IF X(Nfin)<1 THEN 4950
4930 IF (X(I)<=1) AND (X(I+1)>1) THEN Loccne=Lt*12/Nfin*(I+(1-X(I))/(X(I+1)-X(I)))
4940 One(Run)=Locone
4950 Zero(Run)=Loczer
4960 NEXT I
4970 NEXT Run
4980 ASSIGN #1 TO "RE:F8,1"
4990 ASSIGN #2 TO "MDOT:F8,1"
5000 ASSIGN #3 TO "SUPER:F8,1"
5010 ASSIGN #4 TO "ZERO:F8,1"
5020 ASSIGN #5 TO "ONE:F8,1"
5030 MAT PRINT #1,1;Re
5040 MAT PRINT #2,1;Mdot
5050 MAT PRINT #3,1;Super
5060 MAT PRINT #4,1;Zero
5070 MAT PRINT #5,1;One
5080 Omega: END
5090 !
5100 ! THIS SET OF SUB-PROGRAMS CALCULATES THE THERMODYNAMIC AND TRANSPORT PR
OPERTIES OF AIR, GIVEN TEMPERATURE IN DEGREES FAHRENHEIT
5110 !
5120 DEF FNAirtc(T)
5130 !
5140 ! Airtc CALCULATES THE THERMAL CONDUCTIVITY FOR AIR ( BTU/FT.HR./F )
5150 !
5160 Kair=0
5170 A=.012999832
5180 B=2.71167E-5
```

```
5190        C=-1.2811E-8
5200        D=2.0152E-11
5210        E=-3.7245E-14
5220        F=3.5377E-17
5230        G=-1.27737E-20
5240        Kair=A+B*T+C*T^2+D*T^3+E*T^4+F*T^5+G*T^6
5250        RETURN Kair
5260 FNEND
5270 !
5280 DEF FNAirprandtl(T)
5290 !  Airprandtl CALCULATES THE PRANDTL NUMBER OF AIR
5300 !
5310 !
5320        Prair=0
5330        A=.71905671
5340        B=-1.35162E-4
5350        C=-9.1192E-8
5360        D=9.51125E-10
5370        E=-1.56672E-12
5380        F=1.11772E-15
5390        G=-3.0463E-19
5400        Prair=A+B*T+C*T^2+D*T^3+E*T^4+F*T^5+G*T^6
5410        RETURN Prair
5420 FNEND
5430 !
5440 DEF FNAirvisc(T)
5450 !  CALCULATE THE VISCOSITY OF AIR ( LBM/FT.HR. )
5460 !
5470 !
5480        Vis=0
5490        Ta=T+459.69
5500        A=-32.2839
5510        B=.478845
5520        C=5.21402E-4
5530        D=3.96954E-7
5540        E=-1.52477E-10
5550        F=2.27201E-14
5560        Vis=A+B*Ta+C*Ta^2+D*Ta^3+E*Ta^4+F*Ta^5
5570        Muair=3.6E-4*Vis
```

```
5580        RETURN Muair
5590 FNEND
5600  !
5610 DEF FNAircp(T)
5620  !  CALCULATE THE SPECIFIC HEAT FOR FIR ( BTU/LBM /F )
5630  !
5640  !
5650     Cpair=0
5660     Ta=T+459.69
5670        A=.2446604
5680        B=-1.7494E-5
5690        C=-4.632E-9
5700        D=5.5502E-11
5710        E=-3.6663E-14
5720        F=7.1728E-18
5730     Cpair=A+B*Ta+C*Ta^2+D*Ta^3+E*Ta^4+F*Ta^5
5740     RETURN Cpair
5750 FNEND
5760  !
5770  !  THE NEXT TWO SUB-PROGRAMS DETERMINE SATURATION TEMPERATURE AND PRESSUR
E FOR WATER
5780  !
5790 DEF FNSattemp(P)
5800  !
5810  !  CALCULATE THE SATURATION TEMPERATURE (/F) FOR A GIVEN PRESSURE ( psia
)
5820  !
5830     Tsat=0
5840     IF P<=450.0 THEN 5940
5850        V=LOG(P)
5860        A=11545.164
5870        B=-8386.0182
5880        C=2477.7661
5890        D=-363.46271
5900        E=26.690978
5910        F=-.78073813
5920     Tsat=A+B*V+C*V^2+D*V^3+E*V^4+F*V^5
5930     GOTO 6050
5940        V=LOG(10*P)
```

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```
5950            A=35.15789
5960            B=24.592588
5970            C=2.1182069
5980            D=-.3414474
5990            E=.15741642
6000            F=-.0313@9585
6010            G=3.8658@82E-3
6020            H=-2.4901784E-4
6030            I=6.8401@59E-6
6040    Tsat=A+B*V+C*V^2+D*V^3+E*V^4+F*V^5+G*V^6+H*V^7+I*V^8
6050    RETURN Tsat
6060 FNEND
6070 !
6080 DEF FNSatpres(T)
6090 !
6100 !   CALCULATE THE SATURATION PRESSURE ( psia ) FOR A GIVEN TEMPERATURE ( F
6110 !
6120    Psat=0
6130    Tk=(T-32.2)*(5.0/9.0)+273.16
6140    Tc=647.27
6150    Pc=218.167
6160    Pa=14.696
6170    X=Tc-Tk
6180    IF T>=200.0 THEN 6260
6190        A=3.2437814
6200        B=5.86826E-3
6210        C=1.1702379E-8
6220        D=2.1878462E-3
6230        E=(A+B*X+C*X^3)*(1.0/(1.0+D*X))*(X/Tk)
6240        Psat=Pa*Pc/10^E
6250    GOTO 6330
6260        A=3.3463130
6270        B=4.14113E-2
6280        C=7.515484E-9
6290        D=1.3794481E-2
6300        E=6.56444E-11
6310        F=(A+B*X+C*X^3+E*X^4)*(1.0/(1.0+D*X))*(X/Tk)
6320        Psat=Pa*Pc/10^F
```

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```
6330        RETURN Psat
6340    FNEND
6350    !
6360    !   THIS SET OF SUB-PROGRAMS CALCULATES THE THERMODYNAMIC AND TRANSPORT PR
OPERTIES OF WATER AND STEAM ( SUBCOOLED, SFTURATED, AND SUPERHEATED )
6370    !   GIVEN TEMPERATURE IN DEGREEE FAHRENHEIT AND PRESSURE IN psia
6380    !
6390    DEF FNVolcw(P,T)
6400    !
6410    ! CALCULATE THE SPECIFIC VOLUME OF WFTER ( CU.FT./LBM )
6420    !
6430        Spvolw=0
6440        IF P>40.0 THEN 6530
6450            A=.01605514:
6460            B=-2.5908E-6
6470            C=4.3087E-8
6480            D=-1.2311E-10
6490            E=2.0737E-13
6500            F=-2.424E-16
6510            Spvolw=A+B*T+C*T^2+D*T^3+E*T^4+F*T^5
6520        GOTO 6790
6530        IF P>250.0 THEN 6620
6540            A=.01604542
6550            B=-2.4005E-6
6560            C=4.0334E-8
6570            D=-1.0967E-10
6580            E=2.843E-13
6590            F=-4.747E-16
6600            G=4.132E-19
6610        GOTO 6780
6620        IF P>500.0 THEN 6710
6630            A=.01599807
6640            B=-1.0895E-6
6650            C=2.3031E-8
6660            D=3.35E-12
6670            E=-1.217E-13
6680            F=2.933E-16
6690            G=-1.748E-19
6700        GOTO 6780
```

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```
6710            A=.01600488
6720            B=-2.0146E-6
6730            C=3.6511E-8
6740            D=-8.142E-11
6750            E=1.4081E-13
6760            F=-1.1148E-16
6770            G=8.034E-20
6780  Spvolw=A+B*T+C*T^2+D*T^3+E*T^4+F*T^5+G*T^6
6790  RETURN Spvolw
6800  FNEND
6810  !
6820  DEF FNHsl(T)
6830  !
6840  ! CALCULATE THE ENTHALPY OF SATURATED WATER ( BTU/LBM )
6850  Isatw=0
6860  IF T<360.0 THEN 6940
6870            A=-9.041706E2
6880            B=10.673802
6890            C=-4.2753836E-2
6900            D=9.41244E-5
6910            E=-1.0315357E-7
6920            F=4.560246E-11
6930  GOTO 7000
6940            A=-32.179105
6950            B=1.0088084
6960            C=-1.1516996E-4
6970            D=4.8553836E-7
6980            E=-7.3618778E-10
6990            F=9.6350315E-13
7000  Isatw=A+B*T+C*T^2+D*T^3+E*T^4+F*T^5
7010  RETURN Isatw
7020  FNEND
7030  !
7040  DEF FNSsl(T)
7050  !
7060  ! CALCULATE THE ENTROPY OF SATURATED WATER ( BTU/LBM/F )
7070  !
7080  Ssatw=0
7090  IF T<450.0 THEN 7210
```

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```
7100        M=-560.0
7110        N=110.0
7120        A=.76209767
7130        B=.13690825
7140        C=7.5137702E-3
7150        D=5.7828937E-3
7160        E=-1.6168801E-3
7170        F=-2.1403201E-3
7180        G=3.9726934E-3
7190        H=3.4265601E-3
7200        GOTO 7310
7210        M=-360.0
7220        N=310.0
7230        A=.51575516
7240        B=.39679646
7250        C=-4.5979941E-2
7260        D=3.4251697E-2
7270        E=-6.0723333E-3
7280        F=-3.6703358E-3
7290        G=1.2035893E-2
7300        H=1.234655E-2
7310        Tb=(T+M)/N
7320        Ssatw=A+B*Tb+C*Tb^2+D*Tb^3+E*Tb^4+F*Tb^5+G*Tb^6+H*Tb^7
7330        RETURN Ssatw
7340        FNEND
7350        !
7360        DEF FNSphsatw(T)
7370        !
7380        ! CALCULATE THE SPECIFIC HEAT FOR SATURATED WATER ( BTU/LBM F )
7390        !
7400        Cpsatw=0
7410        A=.784791
7420        B=.00525143
7430        C=-4.8694E-5
7440        D=2.1982212E-7
7450        E=-5.26892E-10
7460        F=6.35459E-13
7470        G=-3.078013E-16
7480        Invcpw=A+B*T+C*T^2+D*T^3+E*T^4+F*T^5+G*T^6
```

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```
7490    Cpsatw=1.0/Invcpw
7500    RETURN Cpsatw
7510    FNEND
7520    !
7530    DEF FNSatwprno(T)
7540    !
7550    ! CALCULATE THE PRANDTL NUMBER  FOR SATURATED WATER
7560    !
7570    Prsatw=0
7580        A=-.0490197
7590        B=.005501673
7600        C=-.00006632881
7610        D=.000000051648
7620        E=-1.912988E-9
7630        F=3.600413E-12
7640        G=-3.357675E-15
7650        H=1.213506E-18
7660    Invprw=A+B*T+C*T^2+D*T^3+E*T^4+F*T^5+G*T^6+H*T^7
7670    Prsatw=1.0/Invprw
7680    RETURN Prsatw
7690    FNEND
7700    !
7710    DEF FNSatwtc(T)
7720    !
7730    ! CALCULATE THE THERMAL CONDUCTIVITY FOR SATURATED WATER ( BTU/HR.FT./F )
7740    !
7750    Ksatw=0
7760        A=294.48611
7770        B=1.000526
7780        C=-5.52837E-3
7790        D=2.533937E-5
7800        E=-7.28329E-8
7810        F=1.01215E-10
7820        G=-5.366486E-14
7830    Kw1000=A+B*T+C*T^2+D*T^3+E*T^4+F*T^5+G*T^6
7840    Ksatw=Kw1000/1000.00
7850    RETURN Ksatw
7860    FNEND
7870    !
```

```
7880 DEF FNSatwvisc(T)
7890 !
7900 ! CALCULATE THE VISCOSITY OF SATURATED WATER ( LBM/HR.FT. )
7910 !
7920 IF T>705 THEN T=700
7930 Vis=0
7940 IF T>290.0 THEN 8050
7950    A=72.75867
7960    B=-1.6378164
7970    C=.02140938
7980    D=-1.773353?E-4
7990    E=9.51008E-7
8000    F=-3.19775E-9
8010    G=6.07367E-12
8020    H=-4.9235E-15
8030    Vis=A+B*T+C*T^2+D*T^3+E*T^4+F*T^5+G*T^6+H*T^7
8040    GOTO 8220
8050 IF T>600.0 THEN 8130
8060    A=17.78575
8070    B=-.0963629
8080    C=2.380479E-4
8090    D=-2.779399E-7
8100    E=1.238119E-10
8110    Vis=A+B*T+C*T^2+D*T^3+E*T^4
8120    GOTO 8220
8130 IF T>695.0 THEN 8190
8140    A=-3.35537
8150    B=.02046975
8160    C=-2.023111E-5
8170    Vis=A+B*T+C*T^2
8180    GOTO 8220
8190    A=240.5726596
8200    B=-36.56075447
8210    Vis=A+B*LOG(T)
8220    Musatw=.1158264+Vis
8230    RETURN Musatw
8240 FNEND
8250 !
```

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```
8260  DEF FNSatsvisc(T)
8270  !
8280  ! CALCULATE THE VISCOSITY OF SATURATED STEAM ( LBM/HR.FT. )
8290  !
8300        Vis=0
8310        IF T>400.0 THEN 8380
8320            A=.1826053825
8330            B=2.914147577E-4
8340            C=3.4626801E-7
8350            D=-3.201841?E-10
8360            Vis=A+B*T+C*T^2+D*T^3
8370        GOTO 8560
8380        IF T>645.0 THEN 8460
8390            A=.418496
8400            B=-2.0573E-3
8410            C=9.04919E-6
8420            D=-1.470579E-8
8430            E=8.98275E-12
8440            Vis=A+B*T+C*T^2+D*T^3+E*T^4
8450        GOTO 8560
8460        IF T>700.0 THEN 8530
8470            A=-59.868
8480            B=.288498
8490            C=-4.61343E-4
8500            D=2.46671E-7
8510            Vis=A+B*T+C*T^2+D*T^3
8520        GOTO 8560
8530            A=-155.19044:22
8540            B=23.78556532
8550            Vis=A+B*LOG(T)
8560        Musats=.1158264*Vis
8570        RETURN Musats
8580  FNEND
8590  !
8600  DEF FNSatsprno(T)
8610  !
8620  ! CALCULATES THE PRANDTL NUMBER OF SATURATED STEAM
8630  !
8640        Prsats=0
```

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```
8650       A=1.216696
8660       B=-8.6038E-4
8670       C=-2.7844E-6
8680       D=2.86414E-8
8690       E=-1.15555E-10
8700       F=2.01177E-13
8710       G=-1.283621E-16
8720       Invprs=A+B*T+C*T^2+D*T^3+E*T^4+F*T^5+G*T^6
8730       Prsats=1.00/Invprs
8740       RETURN Prsats
8750 FNEND
8760 !
8770 DEF FNSphsats(T)
8780 !
8790 ! CALCULATE THE SPECIFIC HEAT FOR SATURATED STEAM ( BTU LBM F )
8800 !
8810       Cpsats=0
8820       A=2.3118805
8830       B=-.0018706775
8840       C=1.965058E-5
8850       D=-1.284045E-7
8860       E=3.013665E-10
8870       F=-3.231449E-13
8880       G=1.310507E-16
8890       Invcps=A+B*T+C*T^2+D*T^3+E*T^4+F*T^5+G*T^6
8900       Cpsats=1.00/Invcps
8910       RETURN Cpsats
8920 FNEND
8930 !
8940 DEF FNSatstc(T)
8950 !
8960 ! CALCULATED THE THERMAL CONDUCTIVITY FOR SATURATED STEAM ( BTU/HR.FT./F )
8970 !
8980       Ksats=0
8990       IF T>460.0 THEN 9070
9000       A=14.2703713G6
9010       B=-7.61154325E-2
9020       C=.0006749682
9030       D=-1.8003646E-6
```

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```
9040        E=1.7900443E-9
9050    Ks1000=A+B*T+C*T^2+D*T^3+E*T^4
9060    GOTO 9250
9070    IF T>600.0 THEN 9150
9080        A=429.014
9090        B=-3.70237
9100        C=.01233493
9110        D=-1.809064E-5
9120        E=1.008864E-8
9130    Ks1000=A+B*T+C*T^2+D*T^3+E*T^4
9140    GOTO 9250
9150    IF T>700.0 THEN 9220
9160        A=-16674.5
9170        B=79.4598
9180        C=-.1263192
9190        D=6.72313E-5
9200    Ks1000=A+B*T+C*T^2+D*T^3
9210    GOTO 9250
9220        A=-40486.22807
9230        B=6197.073611
9240    Ks1000=A+B*LOG(T)
9250    Ksats=Ks1000/1000.00
9260    RETURN Ksats
9270    FNEND
9280    !
9290    DEF FNViscs(P,T)
9300    !
9310    ! CALCULATE THE VISCOSITY OF STEAM ( LBM/HR.FT. )
9320    !
9330    Vis=0
9340    IF P>=50.0 THEN 9440
9350        A=1.926629
9360        B=.00002555
9370        C=.0000208279
9380        D=-.00000000049502
9390        E=6.3081E-11
9400        F=-4.0799E-14
9410        G=1.05411E-17
9420    Vis=A+B*T+C*T^2+D*T^3+E*T^4+F*T^5+C*T^6
```

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```
9430        GOTO 9510
9440        A=5.72541
9450        B=-.0332628
9460        C=.00012873E2
9470        D=-2.088647E-7
9480        E=1.636259E-10
9490        F=-4.96576E-14
9500        Vis=A+B*T+C*T^2+D*T^3+E*T^4+F*T^5
9510        Mustm=.01158264*Vis
9520        RETURN Mustm
9530  FNEND
9540  !
9550  SUB Entcw(P,T,Icompw)
9560  !
9570  ! CALCULATE THE ENTHALPY OF SUBCOOLED WATER ( BTU/LBM )
9580  !
9590  OPTION BASE 1
9600  DIM A(5),B(5,5)
9610  MAT A=ZER
9620  Icompw=0
9630  IF P<250.0 THEN 9900
9640        B(1,1)=-3.18831669008E1
9650        B(1,2)=2.98504393234E-3
9660        B(1,3)=1.33348288125E-7
9670        B(1,4)=9.70260583567E-10
9680        B(1,5)=-1.10362284847E-12
9690        B(2,1)=1.00048960506
9700        B(2,2)=-1.05314335283E-5
9710        B(2,3)=1.47662503309E-8
9720        B(2,4)=-3.48793667820E-11
9730        B(2,5)=2.62881336815E-14
9740        B(3,1)=2.82721817375E-5
9750        B(3,2)=-3.12450383809E-7
9760        B(3,3)=9.33318841114E-10
9770        B(3,4)=-1.04460627415E-12
9780        B(3,5)=3.99352883388E-16
9790        B(4,1)=-1.41016204839E-6
9800        B(4,2)=1.11163180927E-8
9810        B(4,3)=-3.06157771954E-11
```

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```
9820    B(4,4)=3.50394655357E-14
9830    B(4,5)=-1.43655502814E-17
9840    B(5,1)=4.07664360366E-10
9850    B(5,2)=-1.87878835824E-13
9860    B(5,3)=1.10163365709E-15
9870    B(5,4)=-2.06875806005E-18
9880    B(5,5)=1.21748558667E-21
9890    GOTO 10420
9900    IF P<15.0 THEN 10170
9910    B(1,1)=-32.298005582
9920    B(1,2)=1.27073877487E-2
9930    B(1,3)=-1.24633564596E-4
9940    B(1,4)=7.14487908312E-7
9950    B(1,5)=-1.36503536452E-9
9960    B(2,1)=1.01381573376
9970    B(2,2)=-2.93828430308E-4
9980    B(2,3)=2.86589395496E-6
9990    B(2,4)=-1.30522857424E-8
10000   B(2,5)=2.11847605407E-11
10010   B(3,1)=-1.91102040795E-4
10020   B(3,2)=4.08786757295E-6
10030   B(3,3)=-4.09161725681E-8
10040   B(3,4)=1.82954852791E-10
10050   B(3,5)=-2.91997554328E-13
10060   B(4,1)=9.31823679019E-7
10070   B(4,2)=-2.39081198404E-8
10080   B(4,3)=2.47927446956E-10
10090   B(4,4)=-1.11965391764E-12
10100   B(4,5)=1.79677751792E-15
10110   B(5,1)=-1.37661172896E-9
10120   B(5,2)=4.68882864067E-11
10130   B(5,3)=-4.84079843213E-13
10140   B(5,4)=2.118231596E-15
10150   B(5,5)=-3.28078007138E-18
10160   GOTO 10420
10170   B(1,1)=-33.9774136563
10180   B(1,2)=1.42585787825
10190   B(1,3)=-.329355699335
10200   B(1,4)=2.87673452024E-2
```

```
10210    B(1,5)=-8.4243794166883E-4
10220    B(2,1)=1.12118345966
10230    B(2,2)=-8.72001344198E-2
10240    B(2,3)=1.99827532858E-2
10250    B(2,4)=-1.73328589779E-3
10260    B(2,5)=5.05202831066E-5
10270    B(3,1)=-2.65590705018E-3
10280    B(3,2)=1.96097311421E-3
10290    B(3,3)=-4.45063321979E-4
10300    B(3,4)=3.83467251626E-5
10310    B(3,5)=-1.11243809796E-6
10320    B(4,1)=2.55004247051E-5
10330    B(4,2)=-1.92085012998E-5
10340    B(4,3)=4.32399525838E-6
10350    B(4,4)=-3.70393732743E-7
10360    B(4,5)=1.07027718423E-8
10370    B(5,1)=-9.12566489903E-8
10380    B(5,2)=6.92372113929E-8
10390    B(5,3)=-1.54789011162E-8
10400    B(5,4)=1.31989972084E-9
10410    B(5,5)=-3.80217009719E-11
10420  FOR I=1 TO 5
10430    FOR J=1 TO 5
10440      K=J-1
10450      A(I)=A(I)+B(I,J)*P^K
10460    NEXT J
10470  NEXT I
10480  FOR I=1 TO 5
10490    K=I-1
10500    Icompw=Icompw+A(I)*T^K
10510  NEXT I
10520  SUBEND
10530  !
10540  SUB Tccw(P,T,Kcompw)
10550  !
10560  ! CALCULATED THE THERMAL CONDUCTIVITY FOR SUBCOOLED WATER ( BTU/HR.FT. F )
10570  !
10580  OPTION BASE 1
10590  DIM A(5),B(5,4)
```

```
10600    MAT A=ZER
10610    K1000=0
10620    IF P>10.0 THEN 10720
10630    A(1)=307.7700899
10640    A(2)=.696656122
10650    A(3)=-.001412111951
10660    FOR I=1 TO 3
10670    K=I-1
10680    K1000=K1000+A(I)*T^K
10690    NEXT I
10700    Kcompw=K1000/1000.0
10710    SUBEXIT
10720        B(1,1)=307.08772221
10730        B(1,2)=.007133552
10740        B(1,3)=-.00001948973
10750        B(1,4)=1.564303E-8
10760        B(2,1)=.7234487569
10770        B(2,2)=-.0001112715317
10780        B(2,3)=6.3925874E-7
10790        B(2,4)=-5.0782358E-10
10800        B(3,1)=-.0017064I0367
10810        B(3,2)=4.163327E-7
10820        B(3,3)=-3.7367227E-9
10830        B(3,4)=3.09641147E-12
10840        B(4,1)=1.008384811E-6
10850        B(4,2)=1.803085129E-9
10860        B(4,3)=1.469999925E-12
10870        B(4,4)=-2.369000954E-15
10880        B(5,1)=-1.286417756E-10
10890        B(5,2)=-6.667660057E-12
10900        B(5,3)=1.273196646E-14
10910        B(5,4)=-7.32465883E-18
10920    FOR I=1 TO 5
10930        FOR J=1 TO 4
10940            K=J-1
10950            A(I)=A(I)+B(I,J)*P^K
10960        NEXT J
10970    NEXT I
10980    FOR I=1 TO 5
```

```
10990        K=I-1
11000        K1000=K1000+A(I)*T^K
11010    NEXT I
11020    Kcompw=K1000/1000.0
11030 SUBEND
11040 !
11050 SUB Sphcw(P,T,Cpcomw)
11060 !
11070 ! CALCULATE THE SPECIFIC HEAT FOR SUICOLLED WATER ( BTU/LBM/F )
11080 !
11090 OPTION BASE 1
11100 Cpcomw=0
11110 DIM A(7)
11120 MAT A=ZER
11130 IF P>60.0 THEN 11250
11140    A(1)=1.0287136
11150    A(2)=-.0010013384
11160    A(3)=.00001238154
11170    A(4)=-7.22385E-8
11180    A(5)=2.09487E-10
11190    A(6)=-2.29792E-13
11200    FOR I=1 TO 6
11210        K=I-1
11220        Cpcomw=Cpcomw+A(I)*T^K
11230    NEXT I
11240    SUBEXIT
11250    A(1)=1.014979
11260    A(2)=-.00054.8746
11270    A(3)=.0000058493
11280    A(4)=-3.07961E-8
11290    A(5)=9.5209E-11
11300    A(6)=-1.4816.1E-13
11310    A(7)=9.692E-17
11320    FOR I=1 TO 7
11330        K=I-1
11340        Cpcomw=Cpcomw+A(I)*T^K
11350    NEXT I
11360 SUBEND
11370 !
```

```
11380  SUB Tcs(P,T,Kstm)
11390  !
11400  ! CALCULATE THE THERMAL CONDUCTIVITY FOR STEAM ( BTU/HR.FT./F )
11410  !
11420     OPTION BASE 1
11430     DIM A(7),B(5,4)
11440     MAT A=ZER
11450     Ks1000=0
11460     IF P>=50.0 THEN 11600
11470        A(1)=9.90499
11480        A(2)=.0139388
11490        A(3)=.000038914
11500        A(4)=-.0000000589
11510        A(5)=7.615E-11
11520        A(6)=-5.164E-14
11530        A(7)=1.3371E-17
11540     FOR I=1 TO 7
11550        K=I-1
11560        Ks1000=Ks1000+A(I)*T^K
11570     NEXT I
11580     Kstm=Ks1000/1000.0
11590     SUBEXIT
11600        B(1,1)=8.7550405
11610        B(1,2)=.096375699
11620        B(1,3)=-.00005487 6232
11630        B(1,4)=4.1416405E-7
11640        B(2,1)=.014332668
11650        B(2,2)=-.000042096669
11660        B(2,3)=2.2221582E-7
11670        B(2,4)=-1.9572826E-9
11680        B(3,1)=.000043096178
11690        B(3,2)=6.4811772E-7
11700        B(3,3)=-1.5156235E-10
11710        B(3,4)=3.2562448E-12
11720        B(4,1)=-4.2130659E-8
11730        B(4,2)=-3.7618844E-10
11740        B(4,3)=-2.1559946E-13
11750        B(4,4)=-2.35229691E-15
11760        B(5,1)=1.77924019E-11
```

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```
11770        B(5,2)=5.45.4874E-14
11780        B(5,3)=2.04.1744E-16
11790        B(5,4)=6.0458155E-19
11800        FOR I=1 TO 5
11810          FOR J=1 TO 4
11820            K=J-1
11830            A(I)=A(I)+B(I,J)*P^K
11840          NEXT J
11850        NEXT I
11860        FOR I=1 TO 5
11870          K=I-1
11880          Ks1000=Ks1000+A(I)*T^K
11890        NEXT I
11900        Kstm=Ks1000/1000.0
11910      SUBEND
11920      !
11930      SUB Sph=(P,T,Cpstm)
11940      !
11950      ! CALCULATE THE SPECIFIC HEAT FOR STEAM ( BTU/LBM/F )
11960      !
11970      OPTION BASE 1
11980      DIM A(7),B(7,5)
11990      MAT A=ZER
12000      MAT B=ZER
12010      Cpstm=0
12020      IF P>=10.0 THEN 12490
12030        B(1,1)=.4222.96265
12040        B(1,2)=.0302.68522
12050        B(1,3)=-.004.854928
12060        B(1,4)=-.0000059748
12070        B(1,5)=.0000445136
12080        B(2,1)=.000181043581
12090        B(2,2)=-.00017032525 17
12100        B(2,3)=-.000013866942
12110        B(2,4)=1.374.36147E-5
12120        B(2,5)=-1.38328347E-6
12130        B(3,1)=-1.62.26512E-7
12140        B(3,2)=-1.36.26817E-7
12150        B(3,3)=4.8560908E-7
```

```
12160    B(3,4)=-1.4?64727E-7
12170    B(3,5)=1.12507264E-8
12180    B(4,1)=-5.69309996E-10
12190    B(4,2)=2.50520512E-9
12200    B(4,3)=-2.19632537E-9
12210    B(4,4)=5.31864156E-10
12220    B(4,5)=-3.83612442E-11
12230    B(5,1)=1.95501625I3E-12
12240    B(5,2)=-5.82315670J9E-12
12250    B(5,3)=4.10144879I54E-12
12260    B(5,4)=-9.18383604222E-13
12270    B(5,5)=6.33?08744846E-14
12280    B(6,1)=-2.01261424352E-15
12290    B(6,2)=5.38?27678341E-15
12300    B(6,3)=-3.47742978501E-15
12310    B(6,4)=7.47?60528987E-16
12320    B(6,5)=-5.01661911137E-17
12330    B(7,1)=7.02997694396E-19
12340    B(7,2)=-1.79132736742E-18
12350    B(7,3)=1.10536956813E-18
12360    B(7,4)=-2.31734765803E-19
12370    B(7,5)=1.52893617959E-20
12380    FOR I=1 TO 7
12390      FOR J=1 TO 5
12400        K=J-1
12410        A(I)=A(I)+B(I,J)*P^K
12420      NEXT J
12430    NEXT I
12440    FOR I=1 TO 5
12450      K=I-1
12460      Cpstm=Cpstm+A(I)*T^K
12470    NEXT I
12480    SUBEXIT
12490    MAT B=ZER(6,4)
12500    IF P>150.0 THEN 12860
12510    B(1,1)=.4716051822
12520    B(1,2)=.0071901422
12530    B(1,3)=.0000029021186
12540    B(1,4)=-1.0947953E-8
```

```
12550        B(2,1)=-.000308842028
12560        B(2,2)=-3.30841469E-5
12570        B(2,3)=-2.56142116E-7
12580        B(2,4)=4.07364494E-10
12590        B(3,1)=1.40514276E-6
12600        B(3,2)=5.510352E-8
12610        B(3,3)=9.0367755E-10
12620        B(3,4)=-2.13368295E-12
12630        B(4,1)=-2.3995704E-9
12640        B(4,2)=-3.32666941E-11
12650        B(4,3)=-1.54501998E-12
12660        B(4,4)=4.3756116E-15
12670        B(5,1)=1.95901818E-12
12680        B(5,2)=-2.7538866E-15
12690        B(5,3)=1.26930177E-15
12700        B(5,4)=-3.9664668E-18
12710        B(6,1)=-6.16157702E-16
12720        B(6,2)=7.09530059E-18
12730        B(6,3)=-4.50706E-19
12740        B(6,4)=1.34272907E-21
12750   FOR I=1 TO 6
12760      FOR J=1 TO 4
12770         K=J-1
12780         A(I)=A(I)+B(I,J)*P^K
12790      NEXT J
12800   NEXT I
12810   FOR I=1 TO 6
12820      K=I-1
12830      Cpstm=Cpstm+A(I)*T^K
12840   NEXT I
12850   SUBEXIT
12860   MAT B=ZER(5,5)
12870        B(1,1)=-3.16454004
12880        B(1,2)=.050468168
12890        B(1,3)=-.000157021974
12900        B(1,4)=2.31892283E-7
12910        B(1,5)=-9.7309759E-11
12920        B(2,1)=.01818816
12930        B(2,2)=-.000025378215
```

```
12940    B(2,3)=8.1211789E-7
12950    B(2,4)=-1.19184066E-9
12960    B(2,5)=5.0417466E-13
12970    B(3,1)=-.000003784538
12980    B(3,2)=4.7562105E-7
12990    B(3,3)=-1.55257254E-9
13000    B(3,4)=2.27009037E-12
13010    B(3,5)=-9.6702875E-16
13020    B(4,1)=2.7766181E-8
13030    B(4,2)=-3.9361743E-10
13040    B(4,3)=1.30491831E-12
13050    B(4,4)=-1.98408444E-15
13060    B(4,5)=8.1631412E-19
13070    B(5,1)=-8.49272664E-12
13080    B(5,2)=1.21404264E-13
13090    B(5,3)=-4.0761977E-16
13100    B(5,4)=5.9420773E-19
13110    B(5,5)=-2.56227204E-22
13120    FOR I=1 TO 5
13130    FOR J=1 TO 5
13140    K=J-1
13150    A(I)=A(I)+B(I,J)*P^K
13160    NEXT J
13170    NEXT I
13180    FOR I=1 TO 5
13190    K=I-1
13200    Cpstm=Cpstm+A(I)*T^K
13210    NEXT I
13220    SUBEND
13230    !
13240    SUB Combstm(Pa,Tfs,Istm,Sstm,Rhostm)
13250    !
13260    ! CALCULATE ENTHALPY, ENTROPY, AND DENSITY FOR STEAM ( BTU/LBM ); ( BTU/LB
M/F ); ( LBM/CU.FT.)
13270    !
13280    OPTION BASE 1
13290    T=Tfs/1.8+255.38
13300    P=Pa/14.6959
13310    B1=2641.62/T*10.0^(80870.0/T^2)
```

```
13320    B2=82.546
13330    B3=162460.0/T
13340    B4=.21828*T
13350    B5=126970.0/T
13360    Bo=1.89-B1
13370    Ba=Bo*P/T^2
13380    Bb=Bo/2*(P/T)^2
13390    Fo=1.89-B1*(3724.20.0/T^2+2)
13400    F=775.596+.63296*T+.00162467*T^2+47.3635*LGT(T)
13410    B6=Bo*B3-2*Fo*(B2-B3)
13420    B7=2*Fo*(B4-B5)-Bo*B5
13430    B=Bo*(1.0+Ba*(B2-B3+Ba*(B4-B5)*Bo*F))
13440    Bet=1.0/T*((Bo-Fo)*P+Bb*(B6+Bb*Bo*(Bo*(B4-B5)-2*B7)))
13450    Vol=.0160185*(4.55504*T/P+B)
13460    Rhostm=1.0/Vol
13470    Istm=F+.043557*(Fo*P+Bb*(-B6+Bo*(B2-B3+2*B7*Bb)))
13480    Sstm=.809691+LGT(T)-.253801*LGT(P)+.00018052*T-11.4276/T-.355579-.024198
3*Bet
13490    SUBEND
13500    !
13510    SUB Viscw(Pres,Temp,U)
13520    !
13530    ! CALCULATE THE VISCOSITY OF SUB-COOLED WATER ( LBM/HR.FT.)
13540    !
13550    OPTION BASE 0
13560    DIM A(4),B(5,5)
13570    Sum1=0
13580    Sum2=0
13590    A(0)=.0181583
13600    A(1)=.0177624
13610    A(2)=.0105287
13620    A(3)=-.00036744
13630    B(0,0)=.501938
13640    B(0,1)=.235622
13650    B(0,2)=-.274637
13660    B(0,3)=.145831
13670    B(0,4)=-.0270448
13680    B(1,0)=.162888
13690    B(1,1)=.789393
```

```
13700 B(1,2)=-.743539
13710 B(1,3)=.263129
13720 B(1,4)=-.0253093
13730 B(2,0)=-.130356
13740 B(2,1)=.673665
13750 B(2,2)=-.959456
13760 B(2,3)=.347247
13770 B(2,4)=-.0267758
13780 B(3,0)=.907919
13790 B(3,1)=1.207552
13800 B(3,2)=-.687343
13810 B(3,3)=.213486
13820 B(3,4)=-.0822904
13830 B(4,0)=.551119
13840 B(4,1)=.0670665
13850 B(4,2)=-.497089
13860 B(4,3)=.100754
13870 B(4,4)=.0602253
13880 B(5,0)=.146543
13890 B(5,1)=-.0843370
13900 B(5,2)=.195286
13910 B(5,3)=-.032932
13920 B(5,4)=-.0202595
13930 T=(Temp-32)/1.8+273.16
13940 Tc=647.27
13950 Rc=317.763
13960 Dummy=FNVolcw(Pres,Temp)
13970 Rho=1/Dummy
13980 R=Rho/2.2046*3.2808^3
13990 FOR K=0 TO 3
14000 Sum1=Sum1+A(K)*(Tc/T)^K
14010 NEXT K
14020 Uo=SQR(T/Tc)/Sum1
14030 FOR I=0 TO 5
14040 FOR J=0 TO 4
14050 Sum2=Sum2+B(I,J)*(Tc/T-1)^I*(R/Rc-1)^J
14060 NEXT J
14070 NEXT I
14080 U=Uo*EXP(R/Rc*Sum2)/1000000*2.2046/3.2808*3600
```

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```
14090     SUBEND
14100 !
14110 SUB Caprate(Twin,Twb,Twout,Pw,Xt,Tsat,Mdotw,Caprtw,Type$)
14120 !
14130 ! CALCULATE WATER-SIDE HEAT CAPACITY RATE ( BTU/HR./F )
14140 !
14150     H2Odif=Tsat-Twout
14160     IF H2Odif>0 THEN 14210
14170     IF Xt>=1.0 THEN 14250
14180     Type$="SATURATEI"
14190     Twout=Tsat
14200 SUBEXIT
14210     CALL Sphcw((Pw),(Twb),Cpw)
14220     Caprtw=Mdotw*Cpw
14230     Type$="SUBCOOLEI"
14240 SUBEXIT
14250     CALL Sphs((Pw),(Twb),Cps)
14260     Caprtw=Mdotw*Cps
14270     Type$="SUPERHEAT"
14280 SUBEND
14290 !
14300 SUB Metals(T,Km,M$)
14310 !
14320 ! TABULAR LISTING OF SELECTED MATERIFL THERMAL CONDUCTIVITIES
14330 !
14340     PRINT ""
14350     PRINT ""
14360     PRINT ""
14370     PRINT ""
14380     PRINT ""
14390     PRINT ""
14400     PRINT "*******************************************************************
          **********"
14410     PRINT TAB(10),"k, THERMAL CONDUCTIVITY (BTU/HR.FT./F) OF VARIOUS METALS
          "
14420     PRINT ""
14430     PRINT TAB(14),"TABLE #";TAB(23),T;": K FOR SOME SELECTED";TAB(49),M$;" M
ATERIALS"
```

```
14440  PRINT "*******************************************************************
*********"
14450  WAIT 500
14460  PRINT TAB(5), "METAL";TAB(34),"68'F";TAB(40),"212'F";TAB(47),"392'F";TAB(
54),"572'F";TAB(61),"752'F";TAB(68),"1112'F";TAB(75),"1472'F"
14470  IF T>1 THEN 14690
14480  PRINT TAB(1), "STEEL ( C MAX.=1.5 % )"
14490  PRINT TAB(3), "CARBON STEEL C=.5 %";TAB(34),"31";TAB(40),"30";TAB(47),"28
";TAB(54),"26";TAB(61),"24";TAB(68),"20";TFB(75),"18"
14500  PRINT TAB(5),"1.0 %";TAB(34),"25";TAB(40),"25";TAB(47),"24";TAB(54),"23"
;TAB(61),"21";TAB(68),"19";TAB(75),"17"
14510  PRINT TAB(5),"1.5 %";TAB(34),"21";TAB(40),"21";TAB(47),"21";TAB(54),"20"
;TAB(61),"19";TAB(68),"18";TAB(75),"16"
14520  PRINT TAB(3),"NICKEL STEEL NI=0 %";TAB(34),"42"
14530  PRINT TAB(5),20 %";TAB(34),"11"
14540  PRINT TAB(5),40 %";TAB(34),"6"
14550  PRINT TAB(5),80 %";TAB(34),"20"
14560  PRINT TAB(3),"INVAR ; NI=36 %";TAB(34),"6.2"
14570  PRINT TAB(3),"CHROME STEEL CR=0 %";TAB(34),"42";TAB(40),"39";TAB(47),"36
";TAB(54),"32";TAB(61),"28";TAB(68),"23";TFB(75),"21"
14580  PRINT TAB(5),"1 %";TAB(34),"35";TAB(40),"32";TAB(47),"30";TAB(54),"27";T
AB(61),"24";TAB(68),"21";TAB(75),"19"
14590  PRINT TAB(5),"2 %";TAB(34),"30";TAB(40),"28";TAB(47),"26";TAB(54),"24";T
AB(61),"22";TAB(68),"19";TAB(75),"18"
14600  PRINT TAB(5),"5 %";TAB(34),"23";TAB(40),"22";TAB(47),"21";TAB(54),"21";T
AB(61),"19";TAB(68),"17";TAB(75),"17"
14610  PRINT TAB(5),"20 %";TAB(34),"13";TFB(40),"13";TAB(47),"13";TAB(54),"13";
TAB(61),"14";TAB(68),"12";TAB(75),"14"
14620  PRINT TAB(3),"CR-NI ( CHROME-NICKEL ):"
14630  PRINT TAB(5),"15 CR, 10 NI";TAB(34),"11"
14640  PRINT TAB(5),"18 CR, 8 NI ( V2A )";TAB(34),"13";TAB(68),"13";TAB(75),"1
0";TAB(54),"11";TAB(61),"11";TAB(34),"9.4";TAB(40),"10";TAB(47),"1
14650  PRINT TAB(5),"20 CR, 15 NI";TAB(34),"8.7"
14660  PRINT TAB(5),"25 CR, 20 NI";TAB(34),"7.4"
14670  INPUT "ENTER A VALUE FOR THE TUBE THERMAL CONDUCTIVITY",Km
14680  SUBEXIT
14690  PRINT TAB(1), "ALUMINUM:"
14700  PRINT TAB(3),"PURE";TAB(34),"118";TAB(40),"119";TAB(47),"124";TAB(54),"1
24";TAB(61),"132";TAB(68),"144"
```

```
14710 PRINT TAB(3), "AL-CU ( DURALUMIN ) 94-96 AL,"
14720 PRINT TAB(5), "3-5 CU, TRACE MG";TAB(34), "95";TAB(40),"105";TAB(47), "112"
14730 PRINT TAB(3), "AL-MG ( HYDRONALIUM )"
14740 PRINT TAB(5), "91-95 AL, 5-9 MG";TAB(34), "65";TAB(40),"73";TAB(47), "82"
14750 PRINT TAB(3), "AL-SI ( SILUMIN ) 87 AL,"
14760 PRINT TAB(5), "13 SI";TAB(34), "95";TAB(40),"101";TAB(47), "107"
14770 PRINT TAB(3), "AL-SI ( ALUSIL ) 78-80 AL,"
14780 PRINT TAB(5), "20-22 SI";TAB(34), "93";TAB(40),"97";TAB(47),"101";TAB(54), "103"
14790 PRINT TAB(3), "AL-MG-SI 97 AL, 1 MG,"
14800 PRINT TAB(5), "1 SI, 1 MN";TAB(34), "102";TAB(40),"109";TAB(47), "118"
14810 PRINT TAB(1), "COPPER :"
14820 PRINT TAB(3), "PURE";TAB(34), "223";TAB(40),"219";TAB(47),"216";TAB(54), "213";TAB(61), "210";TAB(68), "204"
14830 INPUT "ENTER A VALUE FOR THE FIN THERMAL CONDUCTIVITY",Km
14840 SUBEND
```

LIST OF REFERENCES

1. Babcock and Wilcox Co., _Steam, Its Generation and Use_, Babcock and Wilcox Co., New York, 1963.

2. Shields, Carl D., _Boilers: Types, Characteristics, and Functions_, F.W. Dodge Corporation, New York, 1961.

3. Latham, Robert F., _Introduction to Marine Engineering_, U.S. Naval Institute, Anapolis, Maryland, 1955.

4. Croft, Terrel, _Steam Boilers_, McGraw-Hill Book Co., Inc. New York, 1937.

5. Fryling, Glenn R., _Combustion Engineering_, Combustion Engineering Co., Inc. New York, 1966.

6. _Principles of Naval Engineering_, 3rd ed., NAVPERS 10788-B, p. 255, Training Publications Division, Naval Personnel Program Support Activity, Washinton, DC, 1970.

7. Combs, R.M., LCDR, U.S.N., _Waste Heat Recovery Unit Design for Gas Turbine Propulsion Systems_, M.S. Thesis, Naval Postgraduate School, Monterey, California, 1979.

8. Egusa, Tatsuo, _Short Straight Tube Boiler - S.S.T._, Lectures presented at Naval Postgraduate School, Monterey, California, 23-27 June 1980.

9. Holland, F.A., Moores, R.M., Watson, F.A., and Wilkinson, J.K., _Heat Transfer_, American Elsevier Publishing Co., Inc., New York, 1970.

10. Holman, J.P., _Heat Transfer_, 4th ed., McGraw-Hill Book Co., Inc., New York, 1976.

11. Kays, W.M. and London, A.L., _Compact Heat Exchangers_, McGraw-Hill Book Co., Inc., New York, 1964.

12. Weierman, C., Taborek, J., and Marner, W.J., _Comparison of In-Line and Staggered Banks of Tubes with Segmented Fins_, paper presented at the 15th National Heat Transfer Conference, AIChE-ASME, San Francisco, California, 1975.

    Eckert, E.R.G., and Drake, R.M., _Analysis of Heat and Mass Transfer_, McGraw-Hill Book Co., New York, 1972

14. Jakob, M., Heat Transfer and Flow Resistance in Cross Flow of Gases over Tube Banks, Trans. ASMR, vol. 60, p. 384, 1938.

15. Collier, J.G., Convective Boiling and Condensation, McGraw-Hill Book Co., New York, 1972.

16. Sieder, E.N., and Tate, C.E., Heat Transfer and Pressure Drop of Liquids in Tubes, Inc. Eng. Chem., vol. 28, p. 1429, 1936.

17. Dittus, F.W., and Boelter, L.N.K., Heat Transfer in Auto-mobile Radiators of Tubular Type, Publications in Engineering, University of California, p. 443, 1930.

18. Bergles, A.E., and Rohsenow, W.M., The Determination of Forced Convection Surface-Boiling Heat Transfer, J. Heat Trans. vol. 86, pp. 365-372, 1964.

19. Hsu, Y.Y., On the Size Range of Active Nucleation Cavities on a Heating Surface, J. Heat Trans., 843(3), pp. 207-216, 1962.

20. Han, C.Y., and Griffith, P., The Mechanism of Heat Trans-fer in Nucleate Pool Boiling, Part I, Bubble Initiation Growth and Departure, Int. J. Heat Mass Trans. 8(6), pp. 887-904, 1965.

21. Rohsenow, W.M., A Method of Correlating Heat Transfer Data for Surface Boiling of Liquids, Trans., ASME, 74, p. 969, 1952.

22. Rohsenow, W.M., Heat Transfer with Evaporation, University of Michigan Press, p. 101-150, 1953.

23. Sucec, J., Heat Transfer, Simon and Schuster Technical Outline, Simon and Schuster, Inc., New York, 1975.

24. Hsu, Y.Y., and Graham, R.W., Transport Processes in Boiling and Two-Phase Systems, McGraw-Hill Book Co., New York, 1976.

25. Rohsenow, W.M., and Harnett, J/P., Editors, Handbook of Heat Transfer, McGraw-Hill Book Co., New York, 1973.

26. Levy, S., Forced Convection Sbucooled Boiling-Prediction of Vapor Volumetric Fraction, Int. J. Heat Mass Trans., 10, pp. 951-965, 1967.

27. Tong, L.S., Boiling Heat Transfer and Two-Phase Flow, John Wiley and Sons, Inc., New York, 1965.

28. Chen, J., A Correlation for Boiling Heat Transfer to Saturated Fluids in Convective Flow, ASME, 63-HT-34, 1963.

29. Afimiwala, K.A., Interactive Computer Methods for Design Optimization, Ph.D. Thesis. State University of New York, Buffalo, New York, 1976.

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